ACKNOWLEDGEMENTS

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NOMENCLATURE

a	:	Thermal diffusivity of water, m^2/s
A	:	Cross section area of tube, m^2
C_p	:	Specific heat of water, $J/Kg.^{o}C$
D_o	:	Width of flattened tubes, m
e_I	:	Insulation thickness, m
e_p	:	Tube wall thickness, m
F_r	:	Collector heat removal factor
h_i	:	Convective heat transfer coefficient inside of tubes, $w/m^2.^{o}C$
h_w	:	Wind heat transfer coefficient, $w/m^2.^{o}C$
k	:	Thermal conductivity of water, $w/m.^{o}C$
k_I	:	Thermal conductivity of insulation, $w/m.^{o}C$
k_p	:	Thermal conductivity of tubes, $w/m.^{o}C$
L	:	Lenght of tubes, m
$\dot{m}$	:	Mass flow rate of water, Kg/s
N_t	:	Number of tubes
Q	:	Quantity of heat absorbed by collector per period, J
R	:	Inside radius of tubes before flattened, m
Re	:	Reynolds number
Re_o	:	Reynolds number at the maximum continuous flow rate

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S : The rate of incident radiant heat per unit area, w/m^2

t : Time coordinate, s

t_h : Heating time, s

t_p : Pumping time, s

t_t : Time interval of one period, s

T_a : Ambient temperature, ^oC

T_p : Outside tube wall temperature, ^oC

T_1 : Water temperature at coordinate x and time t in free convection, ^oC

T_2 : Water temperature at coordinate x and time t in forced convection, ^oC

ΔT_{on} : Difference between the temperature at the outlet of the collector and that at bottom of the storage tank at which the pump starts to work, ^oC

ΔT_{off} : Difference between the temperature at the outlet of the collector and that at the bottom of the storage tank at which the pump is shut off, ^oC

U_b : Heat loss coefficient from the bottom of collector, $w/m^2.^oC$

U_t : Heat loss coefficient from the top of collector, $w/m^2.^oC$

U_L : Total heat loss coefficient taken at the temperature of water, $w/m^2.^oC$

U_T	:	Total heat loss coefficient taken at the temperature of outside tube wall, $w/m^2.^oC$
v	:	Water velocity, m/s
$\dot{V}$	:	Volume flow rate of water, cm^3/s
x	:	Position coordinate along the lenght of collector, m
α_1	:	$v.t_t/L$
α_2	:	$a.t_t/L^2$
α_3	:	$D_o.t_t.U_L.F_r/c_p.A$
α_4	:	$(S/U_L + T_a - T_{in})/\Delta T_{off}$
η	:	Thermal efficiency of the collector
θ_1	:	Non-dimensional temperature in free convection
θ_2	:	Non-dimensional temperature in forced convection
ξ	:	Non-dimensional position coordinate
ρ	:	Water density, Kg/m^3
τ	:	Non-dimensional time coordinate

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CHAPTER I

INTRODUCTION

The flat solar collector can be used with natural circulation or with forced circulation.

The solar collector heating system is used with natural circulation , using a storage tank located above the collector, with collector and tank connected by a circulation loop. These systems generally lead to low flow rate through the collector with the fluid undergoing a large temperature rise . Because of the low flow rate and high temperature , the system can get boiling phenomenon. However, these systems are reliable, low operating cost and can be operated in remote area where electricity is not available.

The collector heating system is used with forced circulation using a pump to circulate water. The pump suction is connected to the bottom of the storage tank. A temperature control device actuates the on-off operations of the pump. These systems are high in operating cost but commonly used for the heating of buildings.

The collector which was studied in this report was designed by a team of students at Concordia university [1] and won two first awards in water heating by solar energy and in system efficiency (including storage), in the International Student Competition of 33 universities, at New Mexico, 1975. The collector was designed with two glass covers. The tubes of the collector were flattened and painted with black enamel paint. Water was used as circulating fluid.

In 1976, Nabil Nicolas [2] used this solar collector system and performed experiments to find the optimum operating condition. However, the experiments failed to determine the precise quantity of mass flow rate which offers the maximum efficiency.

In this report, the optimum operating condition of the flat solar collector with forced circulation is investigated by a numerical method.

CHAPTER II

DESCRIPTION OF THE WATER HEATING SYSTEM

The water heating system consists of one flat collector, one storage tank, one circulating pump, one temperature control device and one flow control valve. The system is equipped with a temperature recorder to record continuously the inlet and outlet temperature of the collector. The solar radiation is simulated by a lamp having a power of 1600 watts and a peak energy at $\lambda = 1.1\ \mu m$ and installed along the lenght of the collector and parallel to its surface at a distance of 34.5 cm

Figure 1 shows the picture of the water heating system using the flat collector. Figure 2 shows the schematic diagram of the experimental apparatus [2].

The components of the flat collector heating system can be described briefly as follows:

1. FLAT COLLECTOR :

The flat collector is 91.5 cm wide and 122 cm long. It has 42 tubes of 1.27 cm inside diameter and flattened to an oval cross section and parallel to the lenght of the collector. Both ends are welded to two horizontal 2.5 cm inside diameter headers at the top and the bottom of the collector. The tubes are coated with a black enamel paint. The collector is covered with 2 pieces of glass which have an air space of 2.54 cm in between. Glass cover having a thickness

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of 3.2 mm and a transmittance of 87%. The wooden frame at the bottom is well insulated by 4 cm fiber glass. The flat collector is inclined 43° to the horizontal .

2. WATER STORAGE TANK :

Water to be supplied to the collector is stored in an insulated tank in which a constant head is maintained. Tap water from the city water supply circulates continuously through the tank, entering at the bottom and leaving to drain at the top. This continuous flow inside the tank ensures a constant temperature inside the storage tank. This temperature is equal to the temperature of the tap water . The flow rate of tap water can be adjusted.

3. TEMPERATURE CONTROL DEVICE :

A temperature control device controls the ON - OFF operation of the circulating pump. The device bases on the temperature difference ΔT between water at the outlet of collector and water at the bottom of storage tank. Each setting has two operating points: a ΔT_{on} which controls the ON of the pump, and a ΔT_{off} which controls the OFF of the pump. The characteristic of this device is graphically presented by the curve ΔT_{on} vs ΔT_{off} in Figure 3.

4. CIRCULATING PUMP :

Water is pumped through the solar collector by a circulating pump. Water enters at the bottom and leaves at the top of collector. The suction of the pump is connected to the bottom of the storage tank.

5. FLOW CONTROL VALVE :

A valve at the outlet of the collector controls the water flow rate through the collector. Water during the pumping period is collected in a container and weighed accurately. The system is carefully designed to prevent water from flowing through the collec - tor when the pump is stopped.

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CHAPTER III

MATHEMATICAL FORMULATIONS

III.1. FLATTENED TUBE CROSS SECTION AREA, OVERALL HEAT LOSS COEFFICIENT, ACTUAL HEAT ABSORBED BY THE COLLECTOR:

We assume that the tubes of the collector are flattened to the form shown in Figure 4. As shown in Figure 4,

$$D_i = d + 2.r \qquad (a-1)$$

We assume that the inside wall perimeter keeps the same after flattened:

$$2.\pi.R = 2.d + 2.\pi.r \qquad (a-2)$$

We get from equations (a-1) and (a-2) :

$$d = D_i - 2.r$$

and

$$r = (\pi.R - D_i)/(\pi - 2)$$

The cross section area of flattened tubes is calculated by the formula:

$$A = 2.d.r + \pi.r^2 \qquad (a-3)$$

The heat loss at the top consists of the heat loss by free convection from the glass cover and the heat loss due to the reflective radiation of covers and tube surfaces.[7]

An empirical equation for top loss coefficient is developed by

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Klein [3] as follow:

$$U_t = \left[\frac{N}{\left(\frac{344}{T_p}\right) \cdot \left(\frac{T_p - T_a}{N + f}\right)^{0.31}} + \frac{1}{h_w} \right]^{-1} + \frac{\sigma . (T_p^2 + T_a^2)}{\left[E_p + 0.0425 . N . (1 - E_p)\right]^{-1} + \frac{2.N + f - 1}{E_g} - N}$$

Where N:number of glass covers

σ : Stefan-Boltzmann constant = 5.67 10^{-8} $w/m^2 . {}^oK^4$

h_w : wind heat transfer coefficient = 5.7 + 3.8 V [4]

V : wind velocity, m/s

$f = (1 + 0.04\ h_w + 5\ 10^{-4} h_w^2) . (1 \pm 0.058\ N)$

E_g : emittance of glass = 0.78

E_p : emittance of tube = 0.9

T_a : ambient temperature, ^oK

T_p : average outside wall tube temperature, ^oK

The top loss coefficient can be also read by the graph prepared by J.A.Duffie and W.A.Beckman [5] as shown in Figure 5. We see that its value varies between 2.5 and 5.0 $w/m^2 . {}^oC$ for T_p between 10^oC and 110^oC.

It is assumed that the temperature at the surface of the bottom of the collector is equal to ambient temperature. The bottom heat loss coefficient $U_b = k_I/e_I$ where k_I : thermal conductivity of insulation material , e_I : insulation thickness.The bottom heat loss coefficient of this collector is equal to: U_b = 1.125 $w/m^2 . {}^oC$.

The overall heat loss coefficient taken at the outside tube wall temperature : $U_T = U_t + U_b$.

The actual heat absorbed by the collector per unit area, per unit time is equal to the rate of incident radiant heat minus the total heat loss per unit time, per unit area, written as follow:

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$$q = F_r \left[S - U_T \cdot (T_p - T_a) \right] \qquad (1)$$

where S is incident radiant heat, w/m^2. S consists of direct solar radiation and diffuse solar radiation. In this case, the energy source is from a lamp. The lamp power is 1600 watts. We assume that the energy coming to the flat collector is 80% of total source energy. Then, the useful source energy is 1280 watts. The surface of collector is equal to 1.049 m^2. Then, the incident radiant heat per unit area, per unit time is equal to 1220 w/m^2.

F_r is called the collector heat removal factor which defined by J.A.Duffie and W.A.Beckman [5] as:

$$F_r = \frac{\dot{m} \cdot c_p \cdot (T_{wo} - T_{wi})}{N_t \cdot D_o \cdot \Delta x \cdot \left[S - U_T \cdot (T_p - T_a) \right]}$$

where $\dot{m}$: mass flow rate of water

c_p : specific heat of water

Δx : segment along the lenght of tubes

N_t : number of tubes per collector

T_{wo} : water temperature at the end of segment

T_{wi} : water temperature at the entry of segment

D_o : width of flattened tubes.

The collector heat removal factor depends on the geometry of the collector, the mode of heat transfer between tubes and circulating water, the operating temperature. F_r will be determined by experimental data in chapter V.

The actual heat absorbed is written in the conduction term taken through the tubes:

$$q = k_p \cdot (T_i - T_p) / e_p \qquad (2)$$

where T_i is average inside wall tube temperature

k_p is thermal conductivity of tube

e_p is tube thickness

The actual heat absorbed is written in the convection term taken inside tube:

$$q = h_i \cdot (T_i - T) \qquad (3)$$

where h_i : convective heat transfer coefficient inside tube

T : mean water temperature

From the equations (1), (2), (3), we eliminate T_p and T_i and obtain:

$$q = \frac{T_a - T + S/U_T}{\dfrac{1}{F_r \cdot U_T} + \dfrac{e_p}{k_p} + \dfrac{1}{h_i}} \qquad (4)$$

If we call U_L the overall heat loss coefficient calculated with the water temperature, we have:

$$q = F_r \cdot \left[S - U_L \cdot (T - T_a) \right] \qquad (5)$$

From the equations (4), (5), we get:

$$U_L = \frac{S}{T - T_a} + \frac{1 - \dfrac{S}{U_T \cdot (T - T_a)}}{\dfrac{1}{U_T} + \dfrac{F_r \cdot e_p}{k_p} + \dfrac{F_r}{h_i}} \qquad (5\text{-a})$$

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The outlet temperature calculated is not sensitive with the values U_L. Then, the approximative value of U_L can be used. We assume that

-in equation (5-a), F_r is equal to 1 ($F_r \leqslant 1$)

- the mean value of U_t is used. In Figure 5, U_t varies between 2.5 and 5.0 w/m^2.^{o}C. Then, U_t is equal to 3.75 w/m^2.^{o}C.

- $U_T = U_t + U_b$ = 4.88 w/m^2.^{o}C and $(T-T_a)$ is equal to 20^{o}C.

The term $e_p/k_p = 2\ 10^{-6}$ m^2.^{o}C/w may be neglected.

h_i is equal to 300 w/m^2.^{o}C for free convection and 600 w/m^2.^{o}C for forced convection.

We obtain U_L equal to 5.8 w/m^2.^{o}C for free convection and equal to 5.3 w/m^2.^{o}C for forced convection.

III.2. FREE CONVECTION :

ENERGY EQUATION:

We observe the control volume shown in Figure 6. The energy equation can be written as follow:

Rate of gain of energy	=	Rate of energy input by convection	+	Rate of energy input by conduction	+	Rate of energy input by radiation

or

$$\frac{\partial(\rho C_p . T_1)}{\partial t} . A. \Delta x = - \frac{\partial(\rho . v . C_p . T_1)}{\partial x} . A. \Delta x - (-k. \frac{\partial^2 T_1}{\partial x^2}) . A. \Delta x \qquad (6)$$

$$+ D_o . \Delta x.\ F_{r1} . \left[S - U_L . (T_1 - T_a) \right]$$

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where T_1 is the temperature of water at position coordinate x along the lenght of the collector and at time t.

We assume that the convection term being much less significant than the conduction term and the radiation term may be neglected during the heating time. Then, the equation (6) becomes:

$$\frac{\partial T_1}{\partial t} = a.\frac{\partial^2 T_1}{\partial x^2} + \frac{D_o'.F_r.U_L}{\rho.c_p.A}.\left[\frac{\tau S}{U_L} - (T_1 - T_a)\right] \qquad (6\text{-a})$$

where $a = k/\rho.c_p$

BOUNDARY CONDITIONS:

The temperature at the tube inlet is recorded by the temperature recorder. By experimental data, the temperature at the inlet keeps constant at the storage temperature about 800 seconds, then increases linearly with time. The slope of temperature-time curves at the inlet of the collector is measured. Figure 8 shows the variation of slope of temperature-time curves vs control setting temperature. Because the solutions of the problem are not changed significantly with the values of slope, an average value a_o of slope can be used. a_o is equal to 0.011 $^oC/s$.

Then, the boundary condition at the inlet of collector can be expressed by following equation:

$$\text{at } x = 0,\ 0 \leqslant t \leqslant 800\text{seconds},\quad T_1(0,t) = T_{in}$$

$$t > 800 \text{ seconds},\quad T_1(0,t) = a_o.(t-800)+T_{in} \qquad (7)$$

The boundary condition at the outlet of the collector is

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simplified as :

at the end of tube $x = L$, $\frac{\partial T_1(L,t)}{\partial x} = 0$ (8)

INITIAL CONDITION :

At the beginning of each period, $t = 0$, $T_1(x,0) = f(x)$. The function $f(x)$ is unknown. Because the problem was periodical, the temperature distribution along the tube at the end of period must be identical to that at the beginning. Thus,

$$T_1(x,0) = T_2(x, t_t) .$$

Different functions referred to $f(x)$ are given and the respective temperature distributions $T_2(x,t_t)$ are obtained. Figures 9a, 9b, 9c show that $T_2(x,t_t)$ are not sensitive with different functions $f(x)$. Then, the iteration method can be applied to find the initial condition. In effect, it requires only two iterations to get the function $f(x)$ identical to $T_2(x,t_t)$ as shown in Figure 9d.

III.3. FORCED CONVECTION

ENERGY EQUATION :

The pump delivers a mass flow rate $\dot{m}$ equivalent to a mean velocity v or Reynolds number Re.

For forced convection, the equation (6) becomes:

$$\frac{\partial T_2}{\partial t} + v.\frac{\partial T_2}{\partial x} = a.\frac{\partial^2 T_2}{\partial x^2} + \frac{D_o.U_L.F_{r2}}{\rho.c_p.A}.\left[\frac{S}{U_L} - (T_2 - T_a)\right] \quad (10)$$

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BOUNDARY CONDITIONS :

The inlet water temperature decreases to the temperature T_{in} at the bottom of the tank in a very short time .

It is simplified for

$$x = 0 , \quad T_2(o,t) = T_{in} \qquad (11)$$

At the outlet of the tubes, it is assumed that the boundary condition of equation (8) is still held:

$$x = L , \quad \frac{\partial T_2(L,t)}{\partial x} = 0 \qquad (12)$$

INITIAL CONDITION :

After a heating time, t_h , the temperature distribution along the lenght of collector can be written as:

$$t = t_h , \quad T_2(x,t_h) = T_1(x,t_h) \qquad (13)$$

III.4: NON-DIMENSIONAL REPRESENTATION :

The non-dimensional variables corresponding to the position coordinate, time coordinate, temperature are defined as :

$$\xi = \frac{x}{L}$$

$$\tau = \frac{t}{t_t}$$

$$\theta_1 = \frac{T_1 - T_{in}}{\Delta T_{off}}$$

and $$\theta_2 = \frac{T_2 - T_{in}}{\Delta T_{off}}$$

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where t_t : time interval for a period

T_{in}: water temperature at the bottom of the storage tank

ΔT_{off} : setting temperature at which the pump is shut off.

We replace the non-dimensional variables into system of equations (6a) to (13). We get the system of equations in non-dimensional form as follow :

FOR FREE CONVECTION :

EQUATION :

$$\frac{\partial \theta_1}{\partial \tau} = \alpha_2 \cdot \frac{\partial^2 \theta_1}{\partial \xi^2} + \alpha_3 \cdot \left| -\theta_1 + \alpha_4 \right| \quad (6')$$

where $\alpha_2 = a.t_t/L_o^2$

$\alpha_3 = D_o.t_t.U_L.F_{rl}/\rho.C_p.A$

and $\alpha_4 = \left[S/U_L + T_a - T_{in}\right] / \Delta T_{off}$

BOUNDARY EQUATION :

$$\xi = 0 \, , \quad 0 < \tau < \tau_o \, , \quad \theta_1(0,\tau) = 0 \qquad (7')$$

$$\tau_o \leq \tau \, , \quad \theta_1(0,\tau) = a'_o.(\tau - \tau_o)$$

where $a'_o = a_o.t_t/\Delta T_{off}$

$\tau_o = 800/t_t$

$$\xi = 1 \, , \quad \frac{\partial \theta_1(1,\tau)}{\partial \xi} = 0 \qquad (8')$$

INITIAL CONDITION :

$\tau = 0$, $\theta_1(\xi, 0) = g(\xi)$, determined by iterations. (9')

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FOR FORCED CONVECTION :

EQUATION :

$$\frac{\partial \theta_2}{\partial \tau} + \alpha_1 \cdot \frac{\partial \theta_2}{\partial \xi} = \alpha_2 \cdot \frac{\partial^2 \theta_2}{\partial \xi^2} + \alpha_3 \cdot (\alpha_4 - \theta_2) \qquad (10')$$

where $\alpha_1 = v.t_t/L$

BOUNDARY CONDITIONS :

$$\xi = 0 \, , \; \theta_2(0,\tau) = 0 \qquad (11')$$

$$\xi = 1 \, , \frac{\partial \theta_2(1,\tau)}{\partial \xi} = 0 \qquad (12')$$

INITIAL CONDITION :

$$\tau = \tau_h \, , \; \theta_2(\xi,\tau_h) = \theta_1(\xi,\tau_h) \qquad (13')$$

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CHAPTER IV

NUMERICAL ANALYSIS

The heat transfer process is unsteady and assumed one-dimensional . When the variables x and t vary between 0 and L and between 0 and t_t respectively, the variables ξ and τ then vary between 0 and 1. We divide 1 of ξ into (L_h-1) non-dimensional segments and 1 of τ into (K-1) non-dimensional intervals as shown in Figure 7. Then,

one non-dimensional segment is equal to :

$$\Delta\xi = h = 1/(L_h - 1)$$

one non-dimensional interval is equal to :

$$\Delta\tau = k_t = 1/(K-1)$$

the non-dimensional position coordinate is equal to :

$$\xi = (i-1).h \quad \text{where } i = 1, 2, 3, \ldots, L_h-1, L_h$$

and the non-dimensional time coordinate is equal to :

$$\tau = (j-1).k_t \text{ where } j = 1, 2, 3, \ldots, K-1, K$$

The first order partial differential $\frac{\partial\theta}{\partial\xi}$ is written in backward difference as follow :

$$\frac{\partial\theta_{i,j}}{\partial\xi} = \frac{\theta_{i,j} - \theta_{i-1,j}}{h} \tag{14}$$

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CHAPTER IV

NUMERICAL ANALYSIS

The heat transfer process is unsteady and assumed one-dimensional. When the variables x and t vary between 0 and L and between 0 and t_t respectively, the variables ξ and τ then vary between 0 and 1. We divide 1 of ξ into (L_h-1) non-dimensional segments and 1 of τ into (K-1) non-dimensional intervals as shown in Figure 7. Then,

one non-dimensional segment is equal to :

$$\Delta \xi = h = 1/(L_h-1)$$

one non-dimensional interval is equal to :

$$\Delta \tau = k_t = 1/(K-1)$$

the non-dimensional position coordinate is equal to :

$$\xi = (i-1).h \quad \text{where} \quad i = 1, 2, 3 \ldots, L_h-1, L_h$$

and the non-dimensional time coordinate is equal to :

$$\tau = (j-1).k_t \text{ where } j = 1, 2, 3, \ldots\ldots, K-1, K$$

The first order partial differential $\frac{\partial \theta}{\partial \xi}$ is written in backward difference as follow :

$$\frac{\partial \theta_{i,j}}{\partial \xi} = \frac{\theta_{i,j} - \theta_{i-1,j}}{h} \tag{14}$$

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The first order partial differential $\frac{\partial \theta}{\partial \tau}$ is written in backward difference as follow :

$$\frac{\partial \theta}{\partial \tau}_{i,j} = \frac{\theta_{i,j} - \theta_{i,j-1}}{k_t} \quad (15)$$

The second order partial differential $\frac{\partial^2 \theta}{\partial \xi^2}$ is written in finite-difference form as follow :

$$\frac{\partial^2 \theta}{\partial \xi^2}_{i,j} = \frac{\theta_{i+1,j} - 2.\theta_{i,j} + \theta_{i-1,j}}{h^2} \quad (16)$$

We replace the terms (14), (15), (16) into equations (6') to (13') and obtain the difference equations as follows :

FOR FREE CONVECTION :

EQUATION :

$$(1 + 2.k_t.\alpha_2/h^2 + k_t.\alpha_3).\theta_{1_{i,j}} - \theta_{1_{i,j-1}} - (\alpha_2.k_t/h^2).(\theta_{1_{i-1,j}} + \theta_{1_{i+1,j}}) = k_t.\alpha_3.\alpha_4 \quad (6'')$$

$\theta_{i,j-1}$ is known and called C(i). The system of equations (6'') where $i = 2,3,\ldots\ldots,I_h$ forms a tridiagonal matrix.

BOUNDARY CONDITIONS :

$$i = 1, \quad 1 \leqslant j \leqslant K_o, \quad \theta_{1_{i,j}} = 0$$
$$j \geqslant K_o, \quad \theta_{1_{i,j}} = A_o.(j-K_o) \quad (7'')$$

where $K_o=800/t_t.k_t$ and $A_o=a'_o.(\tau_h - \tau_o)/(K_1-K_o)$

$$i = I_h, \quad \theta_{1_{I_h,j}} = \theta_{1_{I_h-1,j}} \quad (8'')$$

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INITIAL CONDITION : It is linearized for 1st iteration :

$$j = 1, \quad \theta_{1\,i,1} = h.(i-1) \qquad (9'')$$

FOR FORCED CONVECTION :

EQUATION :

$$(1 + 2.k_t.\alpha_2/h^2 + k_t.\alpha_3 + k_t.\alpha_1/h).\theta_{2\,i,j} - \theta_{2\,i,j-1} -$$

$$(k_t.\alpha_2/h^2).\theta_{2\,i+1,j} - (k_t.\alpha_2/h^2+k_t.\alpha_1/h).\theta_{2\,i-1,j} = k_t.\alpha_3.\alpha_4 \qquad (10'')$$

Similarly, the system of equations (10'') forms a tridiagonal matrix.

BOUNDARY CONDITIONS :

$$i = 1, \quad \theta_{2\,i,j} = 0 \qquad (11'')$$

$$i = L_h; \quad \theta_{2\,L_h,j} = \theta_{2\,L_h-1,j} \qquad (12'')$$

INITIAL CONDITIONS :

$$j = K_1, \quad \theta_{2\,i,K_1} = \theta_{1\,i,K_1} \qquad (13'')$$

The equations (6'') and (10'') are parabolic. Because the term α_1 is too large, the iteration method is not convergent [6]. The simplest way is to solve the problem by the matrix method. The equations (6'') forms a tridiagonal matrix $\overline{A_1}$ which is the follow:

$$\bar{A}_1 = \begin{array}{c} \left|\begin{matrix} a_1 & -a_3 & & & \\ -a_2 & a_1 & -a_3 & & \\ & -a_2 & a_1 & -a_3 & \\ & & \ddots & \ddots & \ddots \\ & & & a_1 & -a_3 \\ & & & -a_2 & a_1+a_2 \end{matrix}\right| \end{array} \begin{array}{c} 2 \\ \vdots \\ \\ \\ \vdots \\ L_h-1 \end{array}$$

where $a_1 = 1 + 2.k_t.\alpha_2/h^2 + k_t.\alpha_3$

$a_2 = k_t.\alpha_2/h^2$

$a_3 = k_t.\alpha_2/h^2$

Similarly, for forced convection, the tridiagonal matrix is:

$$\bar{A}_2 = \begin{array}{c} \left|\begin{matrix} a_1 & -a_3 & & & \\ -a_2 & a_1 & -a_3 & & \\ & -a_2 & a_1 & -a_3 & \\ & & \ddots & \ddots & \ddots \\ & & & a_1 & -a_3 \\ & & & -a_2 & a_1+a_2 \end{matrix}\right| \end{array} \begin{array}{c} 2 \\ \vdots \\ \\ \\ \vdots \\ L_h-1 \end{array}$$

where $a_1 = 1+2.k_t.\alpha_2/h^2 + k_t.\alpha_3 + k_t.\alpha_1/h$

$a_2 = k_t.\alpha_2/h^2$

$a_3 = k_t.\alpha_2/h^2 + k_t.\alpha_1/h$

The LU decomposition matrix method is applied to solve the problem [7].

The tridiagonal matrix $\overline{A}_1$ can be decomposed into two matrices: the lower matrix $\overline{L}_1$ and the upper matrix $\overline{U}_1$ as follows:

$$\overline{L}_1 = \begin{array}{cc} \begin{bmatrix} 1 & & & & \\ m_3 & 1 & & & \\ & m_4 & 1 & & \\ & & \ddots & \ddots & \\ & & & m_{L_h-1} & 1 \end{bmatrix} & \begin{array}{c} 2 \\ \vdots \\ \\ \vdots \\ L_h-1 \end{array} \end{array}$$

$$\text{and } \overline{U}_1 = \begin{array}{cc} \begin{bmatrix} u_2 & -a_3 & & & \\ & u_3 & -a_3 & & \\ & & \ddots & \ddots & \\ & & & u_{L_h-2} & -a_3 \\ & & & & u_{L_h-1} \end{bmatrix} & \begin{array}{c} 2 \\ \vdots \\ \\ \vdots \\ L_h-1 \end{array} \end{array}$$

where $u_2 = a_1$

$m_i = -a_2/u_{i-1}$ with $u_{i-1} \neq 0$

and $u_i = a_1 + m_i \cdot a_3$ for $i = 3, \ldots, L_h-1$

The system of equations (6'') for $i=2,3,\ldots L_h-1$ can be written in matrix form :

$$\overline{A}_1 \cdot y = b \tag{17}$$

The vector y is $|\theta_{2,j}, \theta_{3,j}, \ldots \theta_{L_h-1,j}|$

The vector b is $|b_2, b_3, b_4, \ldots b_{L_h-1}|$

where $b_2 = \theta_{2,j-1} + (\alpha_2 \cdot k_t/h^2) \cdot \theta_{1,j} + k_t \cdot \alpha_3 \cdot \alpha_4$

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and $b_i = \theta_{i,j-1} + k_t \cdot \alpha_3 \cdot \alpha_4$, for $i = 3,4,\ldots\ldots,I_h-1$

We replace matrices $\overline{L}_1$, $\overline{U}_1$ into equation (17) and obtain :

$$\overline{L}_1 \cdot \overline{U}_1 \cdot y = b \qquad (17')$$

or
$$\overline{L}_1 \cdot z = b \qquad (18)$$

where
$$\overline{U}_1 \cdot y = z \qquad (19)$$

From the equation (18), we obtain the solutions of vector $\underline{z}$ as follows:

$$z_2 = 1,$$

$$z_i = b_i - m_i \cdot z_{i-1} \quad \text{for} \quad i = 3,4,\ldots\ldots,I_h-1.$$

Finally, we obtain the solutions of vector v from the equation (19) as follows:

$$\theta_{I_h-1,j} = z_{I_h-1} / u_{I_h-1}$$

$$\theta_{i,j} = (z_i + a_3 \cdot \theta_{i+1,j})/u_i \quad \text{for} \quad i = I_h-2,\ldots,3,2$$

Similarly, the same procedure of calculations can be applied to find the solutions of system of equations (10'').

The actual energy absorbed is calculated by the trapezoidal rule integration at each step. The formula of the total energy absorbed by the flat solar collector is as follow :

$$Q = \dot{m} \cdot c_p \cdot \Delta T_{off} \cdot k_t \cdot \sum_{j=K_1}^{j=K} \frac{1}{2} \cdot (\theta_{I_h,K_1} + 2 \cdot \theta_{I_h,K_1+1} + \ldots$$

$$\ldots\ldots\ldots + 2 \cdot \theta_{I_h,j} + \ldots\ldots + 2 \cdot \theta_{I_h,K-1} + \theta_{I_h,K})$$

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The thermal efficiency of the collector is calculated by the following formula :

$$\eta = \frac{Q/(t_p+t_h)}{S.W.L}$$

The flow chart of the program is shown in Figure 10.

The program is shown in Figure 11.

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CHAPTER V

RESULTS AND DISCUSSIONS

Four experimental curves of Figures 12,13,14 and 15 obtained from Nicolas' report are used to determine the values of collector heat removal factor F_r. In effect, the factor F_r is a parameter of the system of equations (6''),(7''),..(13''). Then, there is a determined value F_r which will make the solutions of system of equations (6'')...,(13'') agree with the experimental results. Because the system of equations is divided into two parts: free convection and forced convection. There will be one value F_r for each part. To simplify for programming, the value F_r in free convection must satisfy the following condition:

$$\text{at} \quad t = t_{h\ experimental}, \quad T_1(L,t)_{numerical} = T_1(L,t)_{experimental}$$

and the value F_r in forced convection must satisfy the following condition :

at $T_2(L,t)_{numerical} = T_2(L,t)_{experimental}$, the actual heat absorbed by the collector per period calculated must be equal to the experimental one: $(Q/period)_{numerical} = (Q/period)_{experimental}$

As shown in Figure 16, the values F_r are function of the control setting temperature. It can be seen that F_r increases as the control setting temperature decreases and the value F_r in forced convection is higher than that in free convection.

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We choose $F_r' = 0.41$ for free convection and $F_r = 0.72$ for forced convection from the curve of Figure 16. These values correspond with $\Delta T_{on} = 14^{O}C$. The experiment on solar collector with the control setting temperature $\Delta T_{on} = 14^{O}C$ was carried out. The experimental results of outlet temperature vs time are compared with the calculated values as shown in Figure 17. It is confirmed that the calculated results agreed satisfactorily with the experimental ones.

Figures 12,13,14 and 15 also show the variations of the outlet temperature vs time with Reynolds number as parameter.

Figures 18,19,20,21 and 22 show the variations of the thermal efficiency of the collector vs Reynolds number with the control setting temperature ΔT_{on} as parameter.

For one fixed value of the control setting temperature, if Reynolds number is very small, the flow is continuous because the temperature at the outlet of the collector becomes steady at the temperature that difference with the temperature at the bottom of the storage tank is higher than ΔT_{off}. The Reynolds number corresponding to the maximum continuous flow rate is called Re_o.

As shown in Figures 19,20 and 21, the thermal efficiency increases to a maximum and decreases. It can be seen that the thermal efficiency of the collector is maximum at the maximum continuous flow rate. At this point, the efficiency of the

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collector is sensitive with the change of Reynolds number.

When the Reynolds number increases to a value higher than Re_o, the difference between the equilibrium temperature at this flow rate with that at bottom of the storage tank is lower than ΔT_{off}. The pump will be shut off and the flow becomes periodical. The theorical curve of thermal efficiency drops significantly at Re = 100 as can be seen in Figure 21, then increases lightly up to Re = 220, and decreases lightly. The experimental curve of thermal efficiency is a little different from the theorical curve. The thermal efficiency drops significantly at Re_o = 100, then increases much higher than the theorical curve at Re = 300. This disagreement is possibly due to the assumption that U_L remained constant through the period.

Figure 23 shows that the heat absorbed by the collector per period fluctuates with respect to its constant value Q_o at very large Reynolds number and that the pumping time gradually decreases as the Reynolds number increases. The efficiency being function of the rate of the absorbed heat also fluctuates with respect to its constant value.

Figures 24,25 show the variations of the error of the thermal efficiency vs Reynolds number with the control setting temperature ΔT_{on} as parameter. It can be seen that the accuracy is reasonable.

Figure 26 shows the variation of Re_o vs ΔT_{on} and the maximum efficiency. Re_o decreases as the control setting temperature ΔT_{on} increases.

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Figure 27 shows the maximum efficiency as a function of ΔT_{on}. The maximum efficiency is nearly inversely proportional to the control setting temperature ΔT_{on}.

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CHAPTER VI

CONCLUSIONS

In conclusion, the finite-difference method was used to solve the problem of the unsteady and periodical heat transfer process. The overall heat loss coefficient was assumed constant. The collector heat removal factor was determined experimentally and used to calculate the actual heat absorbed by the collector. It is seen that the obtained numerical results are quite reasonable.

It may be concluded that :

The thermal efficiency of the collector increases as the control setting temperature $\Delta T_{on} - \Delta T_{off}$ decreases.

For a fixed control setting temperature, the thermal efficiency of the flat collector is maximum at the maximum continuous flow rate. At this point, the efficiency of the collector is sensitive with the change of the flow rate.

The maximum continuous flow rate increases as the control setting temperature decreases.

This method being general , simple and accurate will be useful to investigate the dynamic performance of the other flat plate solar collectors.

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REFERENCES

1. Lin, Sui and McQueen,H.J , "Solar water heating system", 1975. Concordia university.

2. Nicolas, Nabil, " An experimental investigation of dynamic thermal performance of a flat solar energy collector". Major technical report, 1976, Concordia university.

3. Klein, S.A., " The effects of thermal capacitance of flat plat solar collectors", M.S. Thesis, University of Wisconsin, 1973

4. McAdams, W.C., " Heat transmission", 3rd Edition, New York, Mc Graw Hill, 1954

5. Duffie, J.A., Beckman,W.A., " Solar energy thermal process ", New York, John Wiley & Sons, 1974

6. Ames, W.F. Ed., " Nonlinear partial differential equations ", Academic press, New York, 1967

7. Ralston, Anthony, " A first course in numerical analysis", McGraw Hill Book Company, 1965

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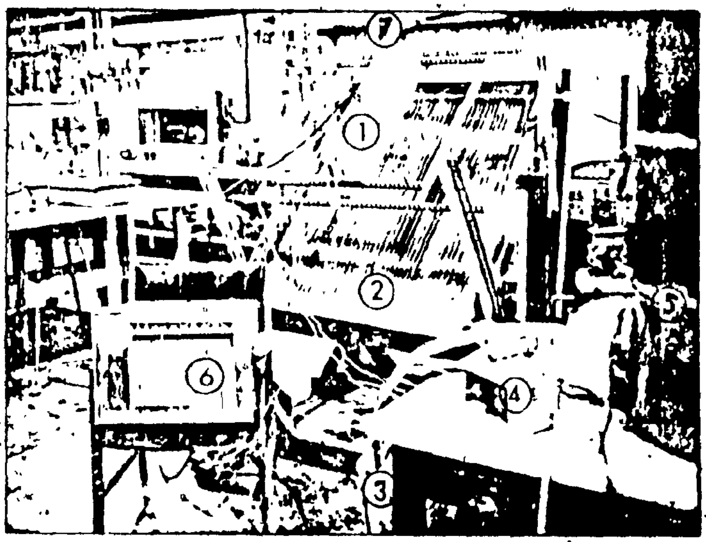

1 Lamp

2 Solar collector

3 Pump

4 Temperature control device

5 Water tank

6 Temp. recorder

7 Flow control valve

Figure 1 APPARATUS

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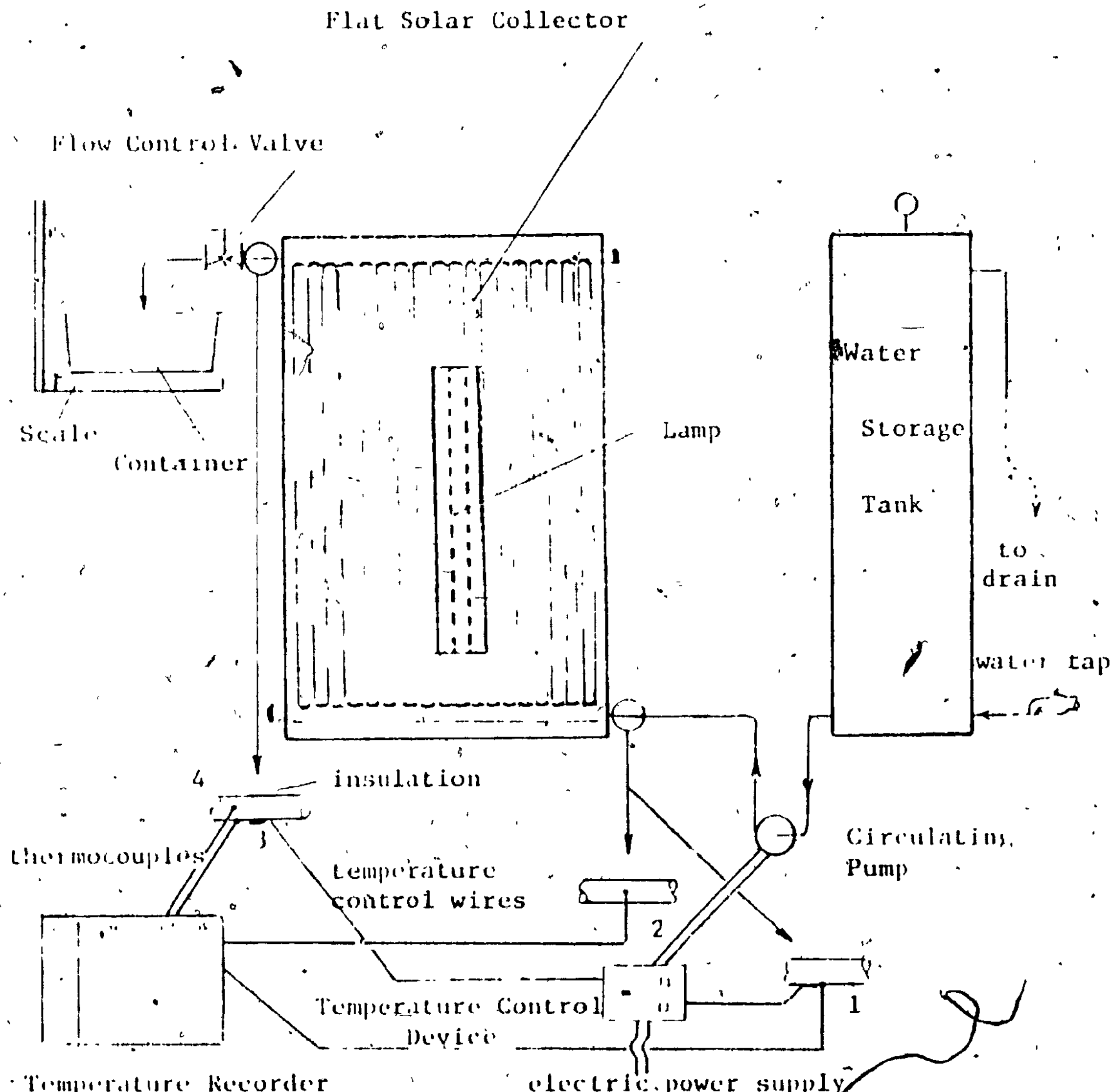

Fig. 2 Schematic diagram of experiment set-up showing solar collector, circulating pump, controls and storage tank.

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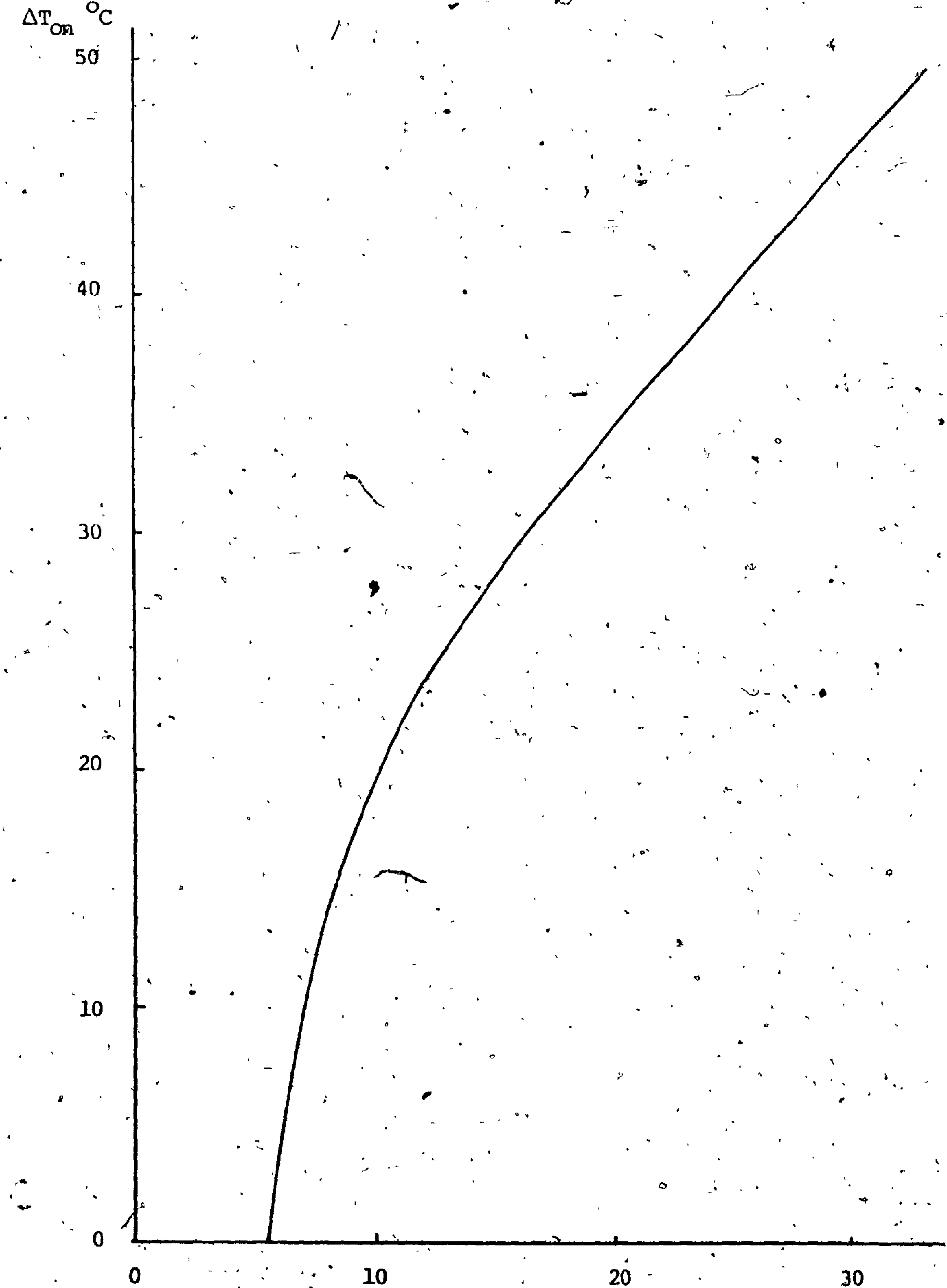

Figure 3. CHARACTERISTICS OF CONTROLLER

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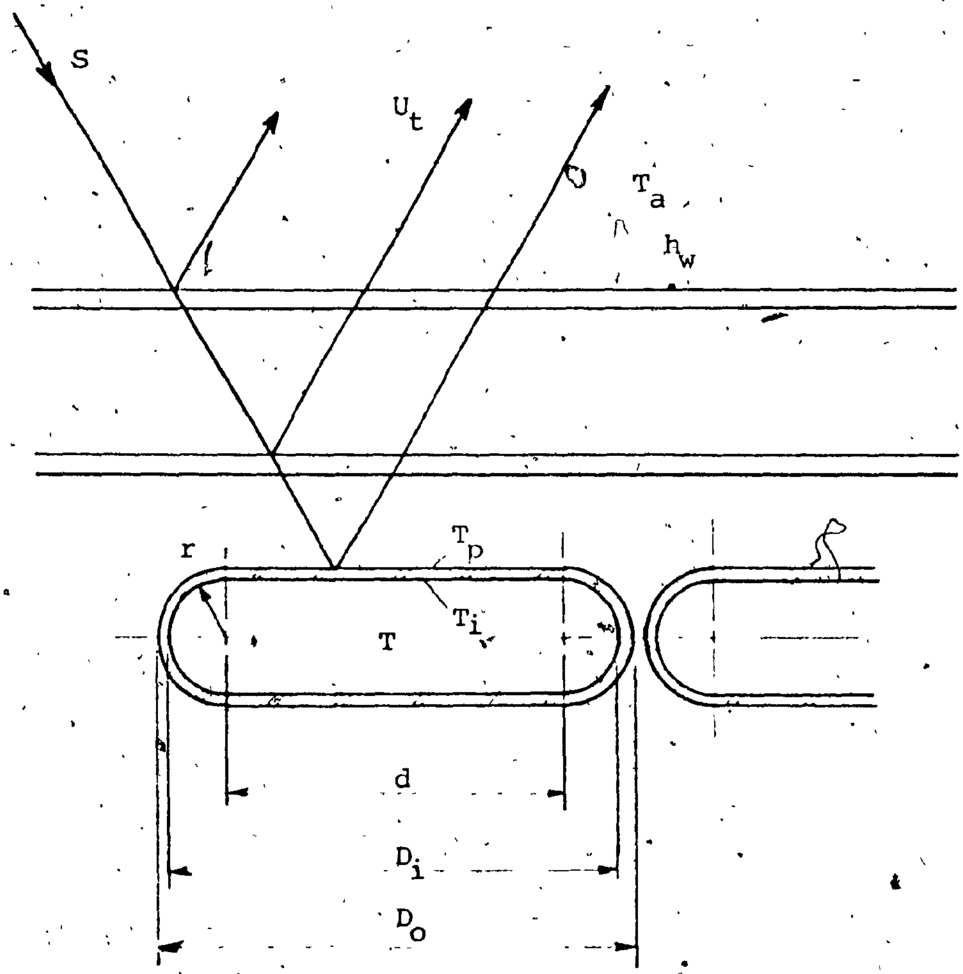

Figure 4. CROSS SECTION OF FLATTENED TUBES OF COLLECTOR

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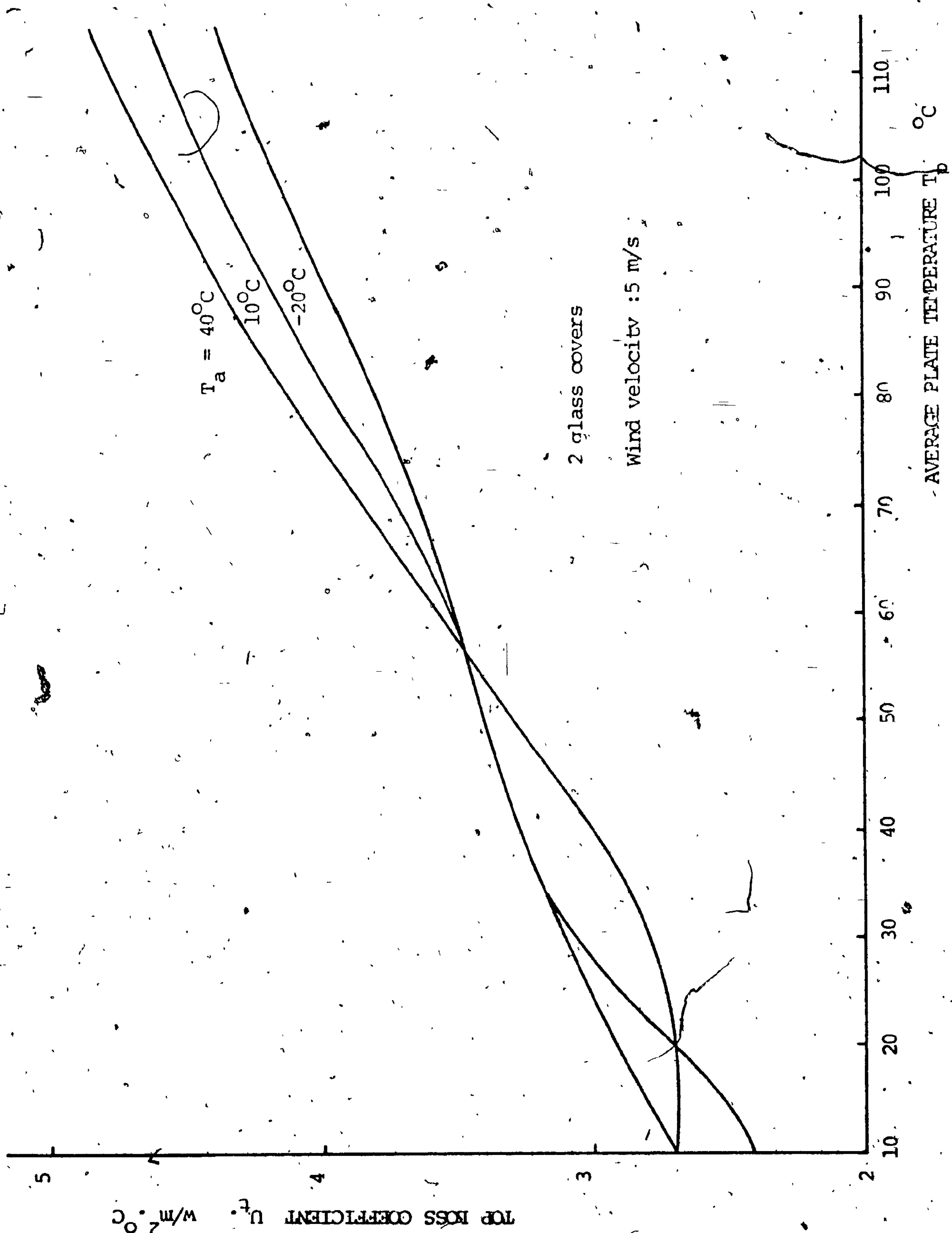

Figure 5

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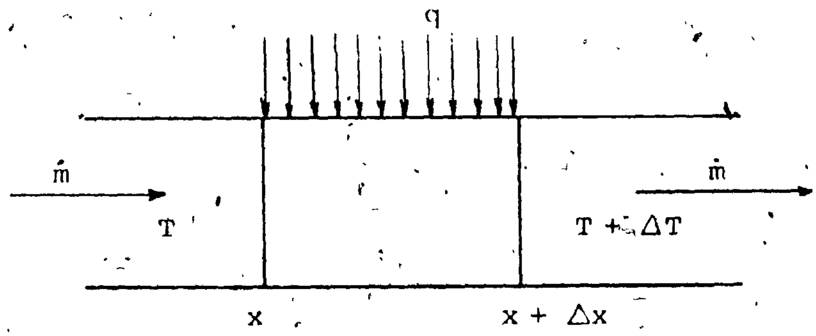

CONTROL VOLUME

FIGURE 6

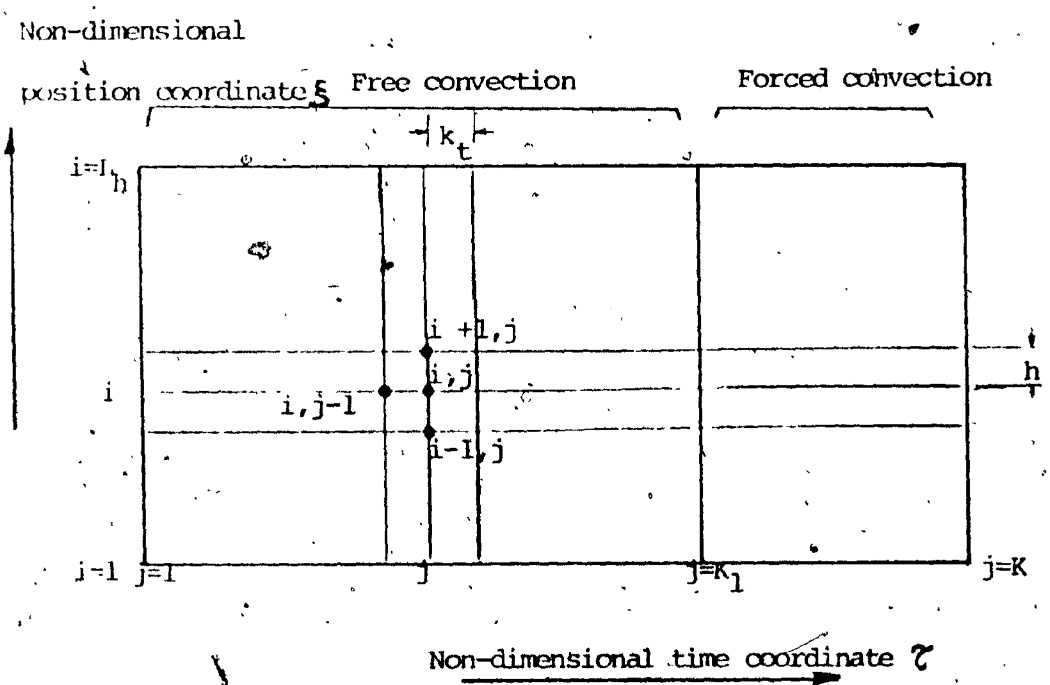

DIAGRAM OF SEGMENTS IN FINITE - DIFFERENCE

Figure 7

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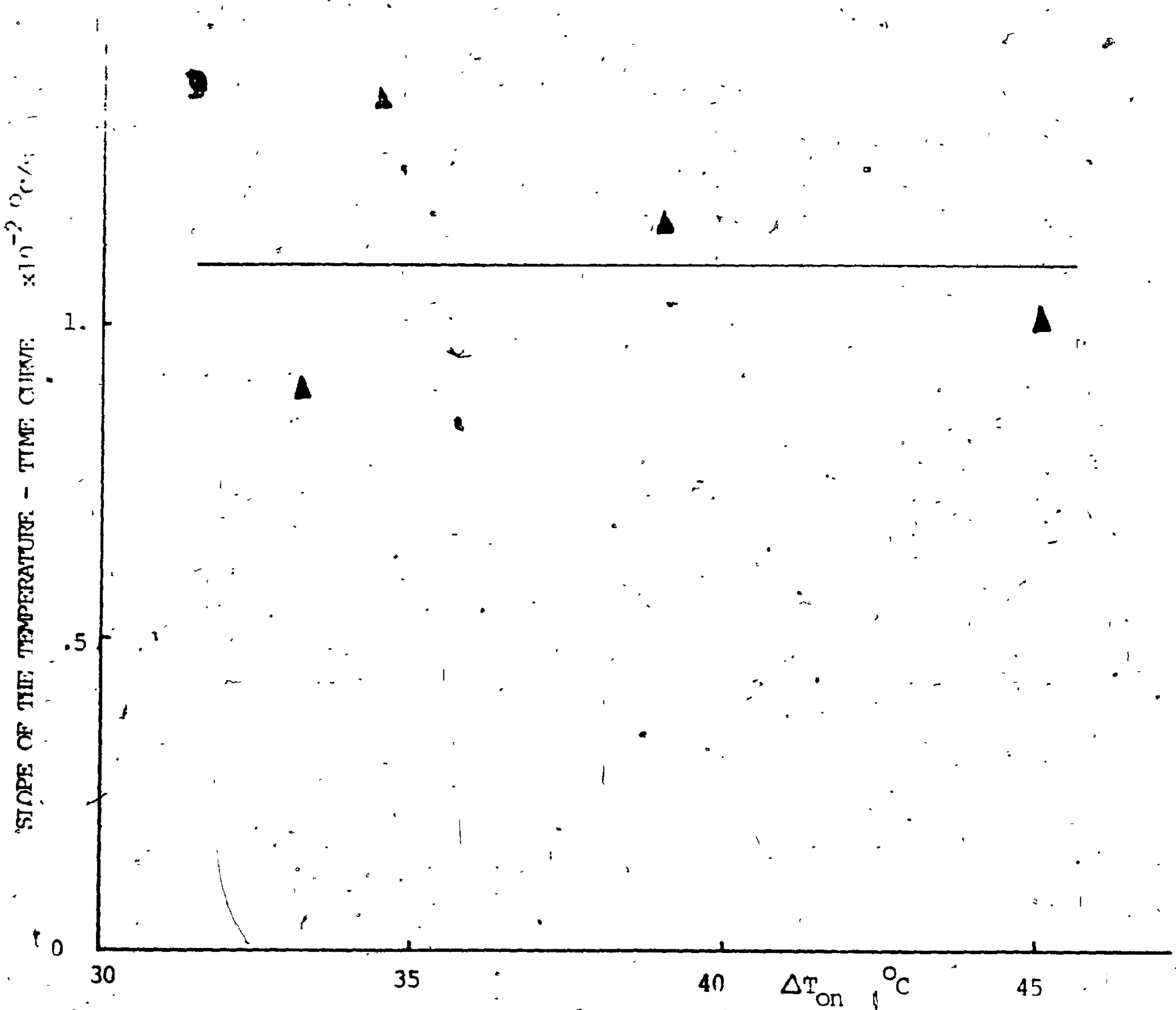

Figure 8. VARIATION OF SLOPE OF THE TEMPERATURE - TIME CURVES VS CONTROL SETTING TEMPERATURE

COMPARISON OF TEMPERATURE DISTRIBUTIONS ALONG THE TUBES BETWEEN THE BEGINNING AND THE END OF PERIOD

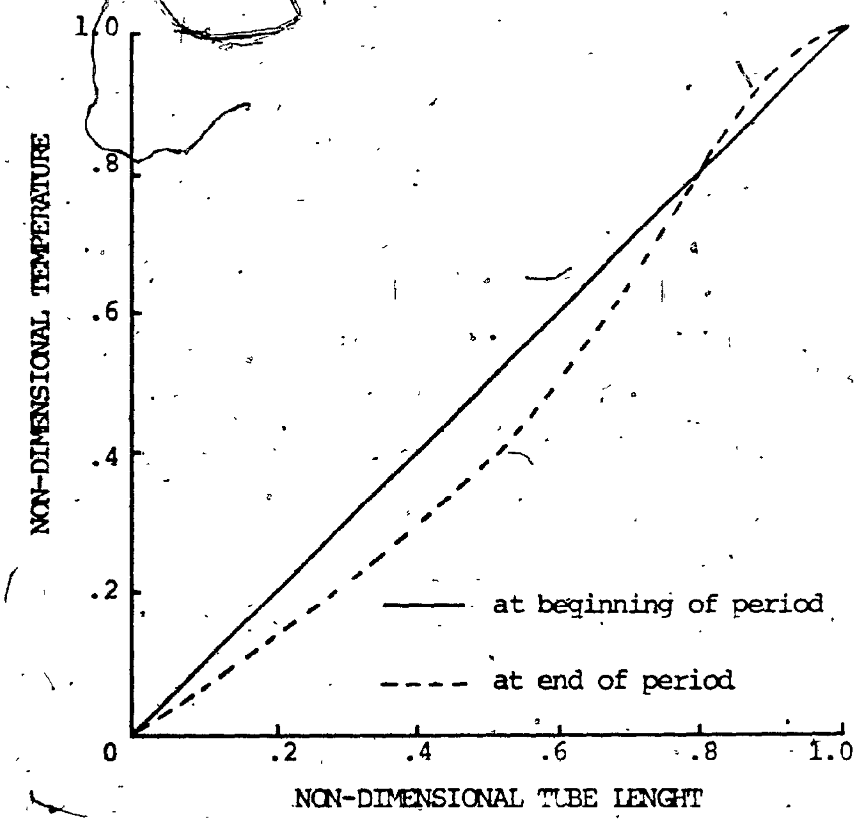

Figure 9a

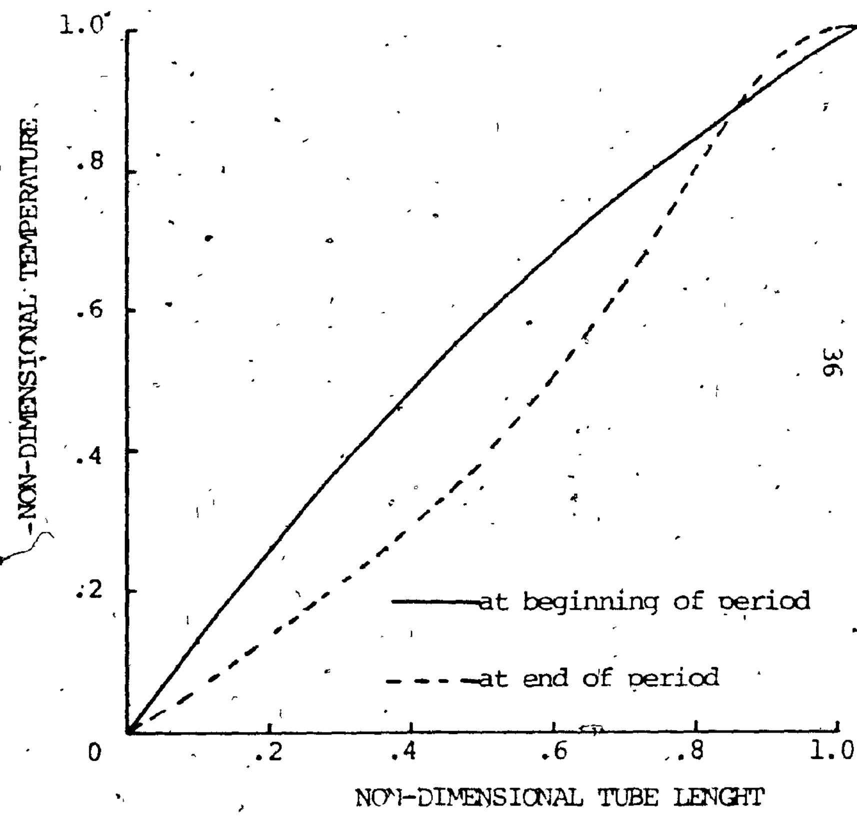

Figure 9b

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COMPARISON OF TEMPERATURE DISTRIBUTIONS ALONG THE TUBES BETWEEN THE BEGINNING AND THE END OF PERIOD

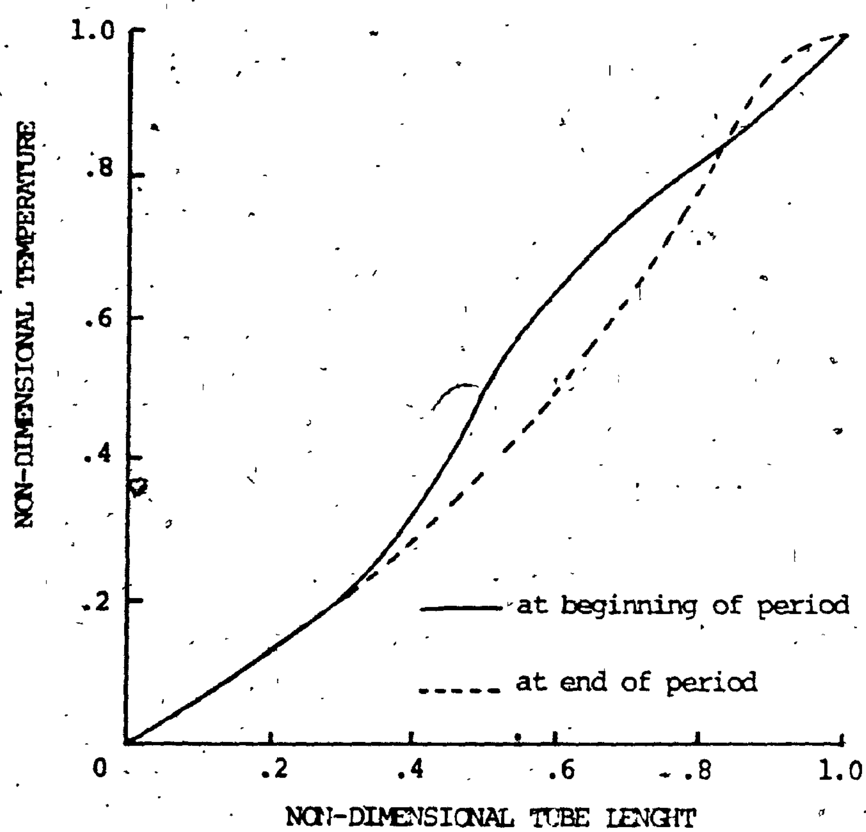

Figure 9c

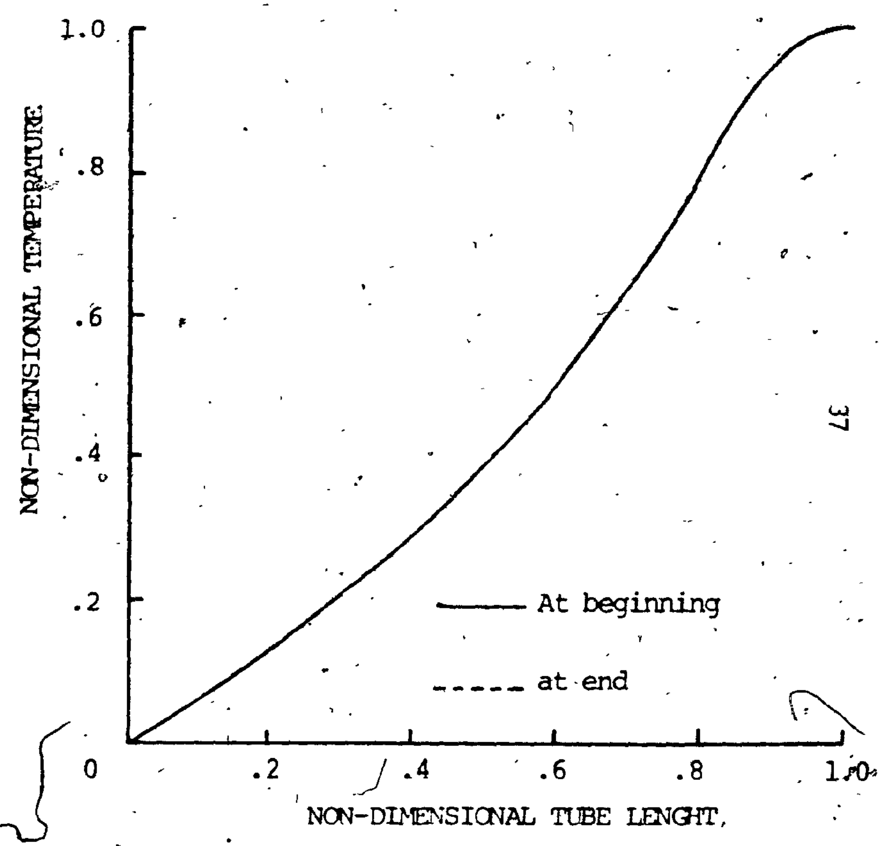

Figure 9d

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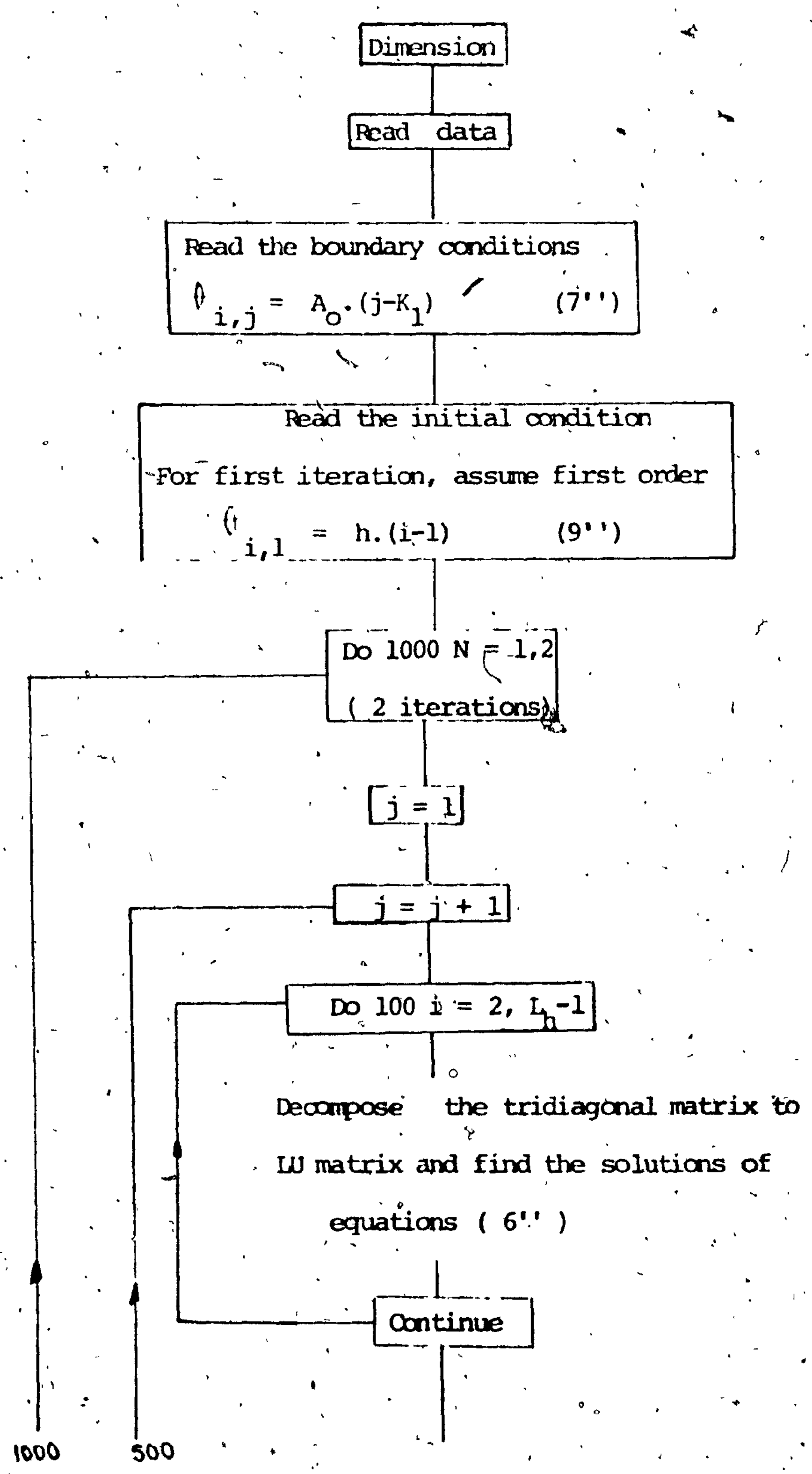
PROGRAM FLOW CHART



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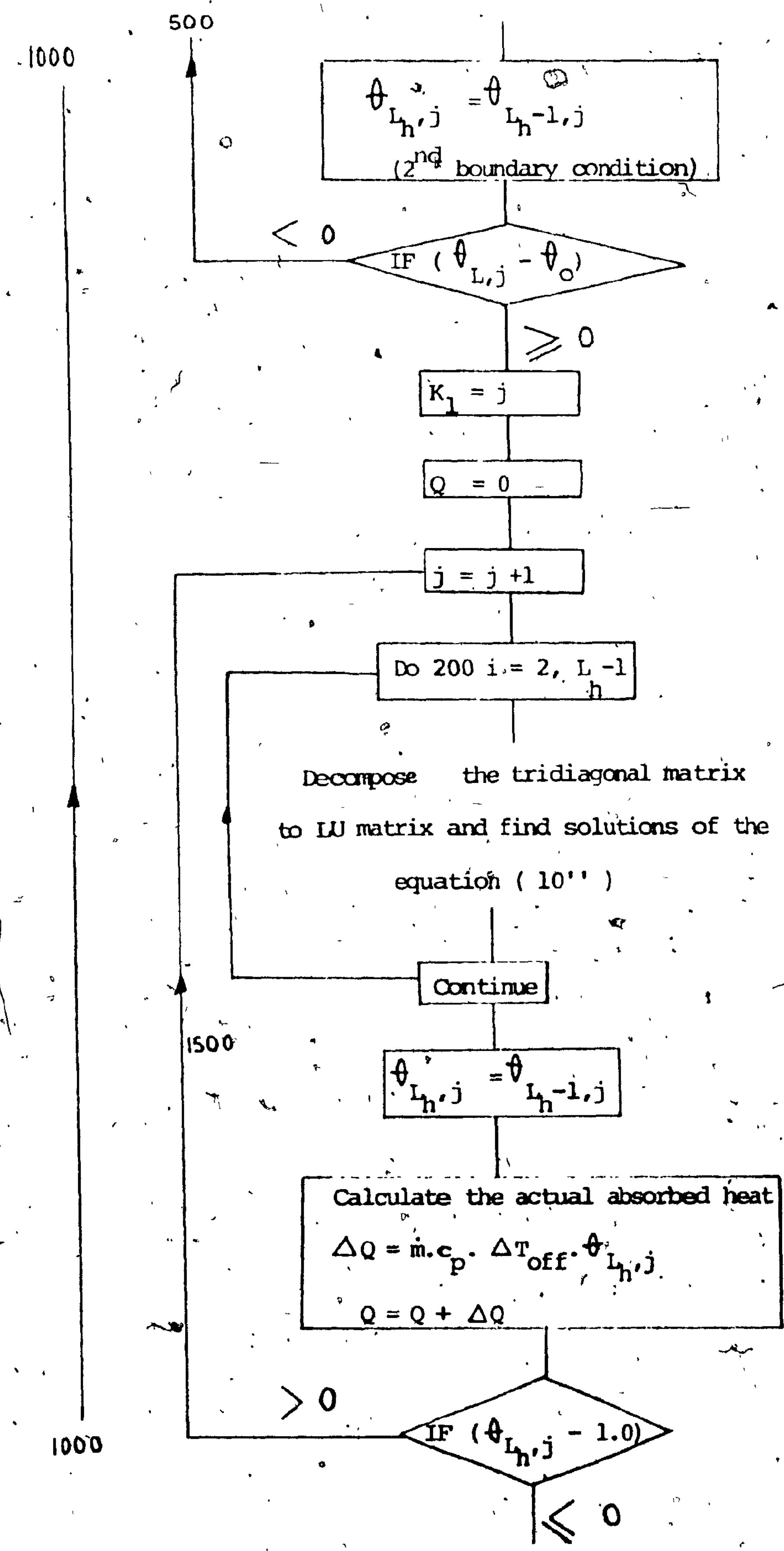

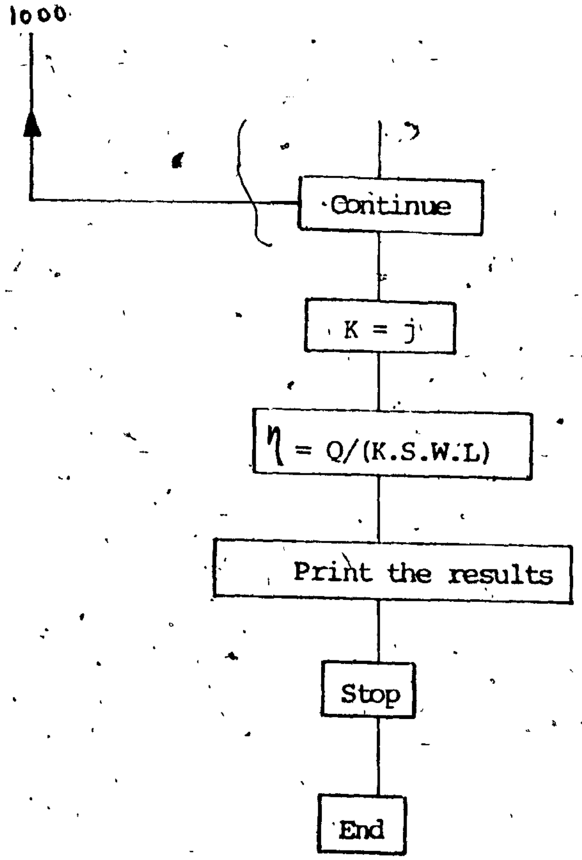
1000
Continue
K = j
η = Q/(K.S.W.L)
Print the results
Stop
End

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```
              PROGRAM DDIB(INPUT,OUTPUT)
          C ****************
000003        DIMENSION D(400),C(20),B(20),BP(20),U(20),TM(20),TD(20),Y(20),X(20
             1),TF(20)
          C ****************
          C  READ DATA
          C
000003        READ 1,L,K,KH,KH2,KH3
000021        READ 2,HSM,GKSM,ALPH1,ALPH2,ALPH3,ALPH4,TFRE,GLEN
          C RX1=X1/X2
000045        READ 16,AREA,CP,X2,S,RX1
000063     16 FORMAT(5F13.7)
000063        X1=RX1*X2
000065        X11=X1
000066        K1=K+1
000070        L1=L-1
000072        L2=L-2
000074        L3=L-3
000076        L4=L-4
000100        KH1=KH+1
000102        KH21=KH2-1
000103        KH31=KH3+1
          C  BOUNDARY CONDITION
000105        DO 210 J=1,KH21
000106    210 D(J)=0.0
000113        DO 220 J=KH2,KH
000115    220 D(J)=D(J-1)+0.0165
000125        DO 230 J=KH1,KH3
000127    230 D(J)=0.0
000134        DO 240 J=KH31,K
000136    240 D(J)=0.0
          C  INITIAL CONDITION
000143        READ 5,SLOP1,SLOP2,SLOP3
          C
          C *********************
000154        DO 5000 NN=1,8
          C ************
000156        PRINT 14,NN
          C ***********
000163        READ 6,ALPH1,ALPH3,ALPH4,ALPH31,ALPH41
          C *************
000201        AMFR=1000.0*AREA*ALPH1*GLEN*42.0/TFRE
          C
000206        C(1)=0.0
000211        DO 10 I=2,4
000213     10 C(I)=C(I-1)+SLOP1*HSM
000224        DO 20 I=5,7
000226     20 C(I)=C(I-1)+SLOP2*HSM
000237        DO 30 I=8,11
000241     30 C(I)=C(I-1)+SLOP3*HSM
          C
          C  END DATA
          C
000252        R1=GKSM*ALPH2/HSM/HSM
000255        R3=ALPH3*GKSM
```

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```
000257          R4=ALPH3*ALPH4*GKSM
000261          R51=1.0+2.0*R1+R3
      C
      C
      C
      C
      C
000266          GO TO (515,515,515,515,515,515,515,515),NN
000302      510 GKSP=GKSM*1.5
000304          GO TO 501
000305      515 GKSP=GKSM
000307          GO TO 501
000307      520 GKSP=GKSM/2.0
000311          GO TO 501
000312      530 GKSP=GKSM/4.0
000314          GO TO 501
000315      540 GKSP=GKSM/6.0
000317          GO TO 501
000320      550 GKSP=GKSM/7.0
000322          GO TO 501
000323      560 GKSP=GKSM/8.0
000325          GO TO 501
000326      570 GKSP=GKSM/12.0
000330          GO TO 501
000331      580 GKSP=GKSM/15.0
000333          GO TO 501
000334      590 GKSP=GKSM/20.0
000336          GO TO 501
000337      501 RP1=GKSP*ALPH2/HSM/HSM
000342          RP2=GKSP*ALPH1/HSM
000344          RP3=ALPH31*GKSP
000346          RP4=RP3*ALPH41
000350          R5=1.0+2.0*RP1+RP3+RP2
      C
      C FOR FREE CONVECTION
      C
000354          A1=R1/R51
000356          A2=1.0/R51
000360          A3=R1/R51
000361          A0=R4/R51
000363          J=1
000364          TH=0.0
000365      450 J=J+1
000367          Y1=GKSM*TFRF
000371          TH=TH+Y1
000373          IF(J-KH2)460,470,470
000376      470 D(J)=D(J-1)+0.0165
000404          GO TO 460
000405      460 DO 400 I=2,L1
000407      400 TD(I)=-A3
000414          BP(L1)=A1*D(J)+A2*C(2)+A0
000426          DO 500 I=1,L2
000430          I1=L1-I
000431          I3=I+1
000433          BP(I1)=A2*C(I3+1)+A0
000442      500 CONTINUE
```

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```
00444          TF(1)=-(A1+A3)
00451          DO 600 I=2,L2
00453     600  TF(I)=-A1
00460          U(1)=1.0
00463          Y(1)=BP(1)
00470          DO 700 I=1,L2
00471          I3=I+1
00473          TM(I3)=TD(I3)/U(I)
00502          U(I3)=1.0-TM(I3)*TF(I)
00512          Y(I3)=BP(I3)-TM(I3)*Y(I)
00524     700  CONTINUE
00527          DO 800 I=1,L1
00530          I1=L-I
00531          I2=I1+1
00533          I3=I+1
00534          IF(I-1)301,301,302
00536     302  X(I1)=(Y(I1)-TF(I1)*X(I2))/U(I1)
00554          GO TO 303
00554     301  X(I1)=Y(I1)/U(I1)
00563          GO TO 303
00564     303  C(I3)=X(I1)
00571     800  CONTINUE
00574          C(1)=D(J)
00601          C2=ABS(C(L)-RX1)
00611          IF(C2-0.05)440,440,450
00614     440  KH=J
00616          GO TO 430
          C
          C
          C FOR FORCED CONVECTION
          C
00616     430  A1=(RP1+RP2)/R5
00622          A2=1.0/R5
00623          A3=RP1/R5
00625          A0=RP4/R5
00626          W=0.0
00627          TP=0.0
00630          SI=-HSM
00632          DO 4100 I=1,L
00633          SI=SI+HSM
00635          FACT=1.0+0.60*SI*(-SI+1.0)
00642          C(I)=C(I)*FACT
00647    4100  CONTINUE
00652          X10=C(L)
00655     420  J=J+1
00657          Y2=GKSP*TFRE
00661          TP=TP+Y2
00663          D(J)=0.0
00666          DO 100 I=2,L2
00670     100  TD(I)=-A1
00675          B(1)=A1*D(J)+A2*C(2)+A0
00707          U(1)=1.0
00712          Y(1)=B(1)
00717          DO 200 I=1,L4
00720          I3=I+1
00722          B(I3)=A2*C(I3+1)+A0
```

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```
00730          TM(I3)=TD(I3)/U(I)
00737          U(I3)=1.0+TM(I3)*A3
00746          Y(I3)=B(I3)-TM(I3)*Y(I)
00760      200 CONTINUE
00763          B(L2)=A2*C(L1)+A0
00771          TM(L2)=TD(L2)/U(L3)
01000          U(L2)=1.0-A3+TM(L2)*A3
01007          Y(L2)=B(L2)-TM(L2)*Y(L3)
01021          DO 300 I=1,L2
01023          I1=L1-I
01024          I2=I1+1
01026          IF(I-1)170,170,180
01030      170 X(I1)=Y(I1)/U(I1)
01037          GO TO 190
01040      180 X(I1)=(A3*X(I2)+Y(I1))/U(I1)
01054          GO TO 190
01054      190 C(I2)=X(I1)
01061      300 CONTINUE
01064          X(L1)=X(L2)
01070          C(L)=X(L1)
01075          C(1)=D(J)
01102          DQ=AMFR*CP*X2*(X10+C(L))*Y2/2.0
01113          W=W+DQ
01115          Z=ABS(X10-C(L))
01125          IF(Z-0.000001) 610,610,620
01130      610 W1=AMFR*CP*X2*C(L)
01135          TPER=TH+TP
01137          X1=C(L)*X2
01143          PRINT 23
01146          PRINT 24,X1
01154          GO TO 630
01155      620 X10=C(L)
01160          C1=ABS(C(L)-1.0)
01170          IF(C1-0.05)410,410,420
01173      410 TPER=TH+TP
01175          W1=W/TPER
01177      630 EFFIC=100.0*W1/S
         C
         C
01202          PRINT 22,X11,X2
01211          PRINT 18,GKSM,GKSP,RX1,X1,X2
01227          PRINT 15,ALPH1,AMFR
01237          PRINT 11,J,KH,TFRE
01251          PRINT 19,TH,TP,TPER
01263          PRINT 17,S,W1,EFFIC
01275     5000 CONTINUE
         C
         C
01277        1 FORMAT(5I4)
01277        2 FORMAT(5F12.10,/,3F14.9)
01277        3 FORMAT(3X,5F13.2)
01277        4 FORMAT(6X,3HI =,I2,4X,E13.4)
01277        5 FORMAT(3F10.7)
01277        6 FORMAT(5F12.6)
01277        7 FORMAT(10X,6HITE = ,I6)
01277        9 FORMAT(2(2X,6E11.4))
```

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```
001277        11 FORMAT(10X,4HK = ,I5,5X,5HKH = ,I5,5X,17HTIME REFERENCE = ,F10.3)
001277        13 FORMAT(2I4)
001277        14 FORMAT(//,10X,5HNN = ,I3,//)
001277        15 FORMAT(10X,8HALPHI = ,F12.4,5X,14HM.FLOW RATE = ,F12.8)
001277        17 FORMAT(10X,4HS = ,F10.3,4HW = ,F10.3,/,10X,13HEFFICIENCY = ,F10.3,
             18HPER CENT)
001277        18 FORMAT(2X,7HGKSM = ,F10.5,2X,7HGKSP = ,F10.5,2X,6HRX1 = ,F10.5,2X,
             15HX1 = ,F10.5,2X,5HX2 = ,F10.5)
001277        19 FORMAT(10X,5HTH = ,F10.2,4HSEC.,5X,5HTP = ,F10.2,4HSEC.,5X,5HTPER=
             1,F10.2,4HSEC.)
001277        21 FORMAT(10X,I5)
001277        22 FORMAT(10X,9HDT OFF = ,F10.3,/,10X,8HDT ON = ,F10.3)
001277        23 FORMAT(10X,27HPUMP   WORKS   CONTINUOUSLY)
001277        24 FORMAT(10X,17HAT TEMPERATURE = ,F8.2)
          C
001277           STOP
001301           END


          NN =    1


          PUMP   WORKS   CONTINUOUSLY
          AT TEMPERATURE =     70.20
          DT OFF =      47.500
          DT ON =       19.000
   GKSM =     .01250   GKSP =      .01250  RX1 =      2.50000  X1 =    70.19869  X2 =
          ALPHI =          1.0000      M.FLOW RATE =      .00229047
          K =    280      KH =     51       TIME REFERENCE =    3172.000
          TH =      1982.50SEC.      TP =      9079.85SEC.      TPER=  11062.35SEC.
          S =    1280.000W =       673.026
          EFFICIENCY =        52.580PER CENT


          NN =    2


          PUMP   WORKS   CONTINUOUSLY
          AT TEMPERATURE =     54.81
          DT OFF =      47.500
          DT ON =       19.000
   GKSM =     .01250   GKSP =      .01250  RX1 =      2.50000  X1 =    54.81106  X2 =
          ALPHI =          1.3097      M.FLOW RATE =      .00299983
          K =    232      KH =     51       TIME REFERENCE =    3172.000
          TH =      1982.50SEC.      TP =      7176.65SEC.      TPER=   9159.15SEC.
          S =    1280.000W =       688.245
          EFFICIENCY =        53.769PER CENT


          NN =    3


          PUMP   WORKS   CONTINUOUSLY
          AT TEMPERATURE =     41.86
          DT OFF =      47.500
          DT ON =       19.000
   GKSM =     .01250   GKSP =      .01250  RX1 =      2.50000  X1 =    41.86358  X2 =
          ALPHI =          1.7463      M.FLOW RATE =      .00399985
          K =    193      KH =     51       TIME REFERENCE =    3172.000
          TH =      1982.50SEC.      TP =      5630.30SEC.      TPER=   7612.80SEC.
          S =    1280.000W =
          EFFICIENCY =        54.758PER CENT
```

MARKS ON ORIGINAL

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        PUMP   WORKS   CONTINUOUSLY
        AT TEMPFRATURE =     28.43
        DT OFF =       47.500
        DT ON =        19.000
GKSM =     .01250   GKSP =       .01250  RX1 =       2.50000  X1 =    28.42736  X2 =
        ALPHI =         2.6194      M.FLOW RATE =      .00599966
        K =    151      KH =     51      TIME REFERENCE =     3172.000
        TH =     1982.50SEC.      TP =     3965.00SEC.      TPER=   5947.50SEC.
        S =    1280.000W =      713.906
        EFFICIENCY =       55.774PER CENT

        NN =    5

        PUMP   WORKS   CONTINUOUSLY
        AT TEMPFRATURE =     24.50
        DT OFF =       47.500
        DT ON =        19.000
GKSM =     .01250   GKSP =       .01250  RX1 =       2.50000  X1 =    24.49585  X2 =
        ALPHI =         3.0559      M.FLOW RATE =      .00699945
        K =    139      KH =     51      TIME REFERENCE =     3172.000
        TH =     1982.50SEC.      TP =     3489.20SEC.      TPER=   5471.70SEC.
        S =    1280.000W =      717.686
        EFFICIENCY =       56.069PER CENT

        NN =    6

        PUMP   WORKS   CONTINUOUSLY
        AT TEMPFRATURE =     21.52
        DT OFF =       47.500
        DT ON =        19.000
GKSM =     .01250   GKSP =       .01250  RX1 =       2.50000  X1 =    21.51886  X2 =
        ALPHI =         3.4925      M.FLOW RATE =      .00799947
        K =    130      KH =     51      TIME REFERENCE =     3172.000
        TH =     1982.50SEC.      TP =     3132.35SEC.      TPER=   5114.85SEC.
        S =    1280.000W =      720.541
        EFFICIENCY =       56.292PER CENT

        NN =    7

        DT OFF =       47.500
        DT ON =        19.000
GKSM =     .01250   GKSP =       .01250  RX1 =       2.50000  X1 =    21.51886  X2 =
        ALPHI =         3.9291      M.FLOW RATE =      .00899948
        K =     87      KH =     51      TIME REFERENCE =     3172.000
        TH =     1982.50SEC.      TP =     1427.40SEC.      TPER=   3409.90SEC.
        S =    1280.000W =      556.229
        EFFICIENCY =       43.455PER CENT

        NN =    8

        DT OFF =       47.500
        DT ON =        19.000
GKSM =     .01250   GKSP =       .01250
```

MARKS ON ORIGINAL

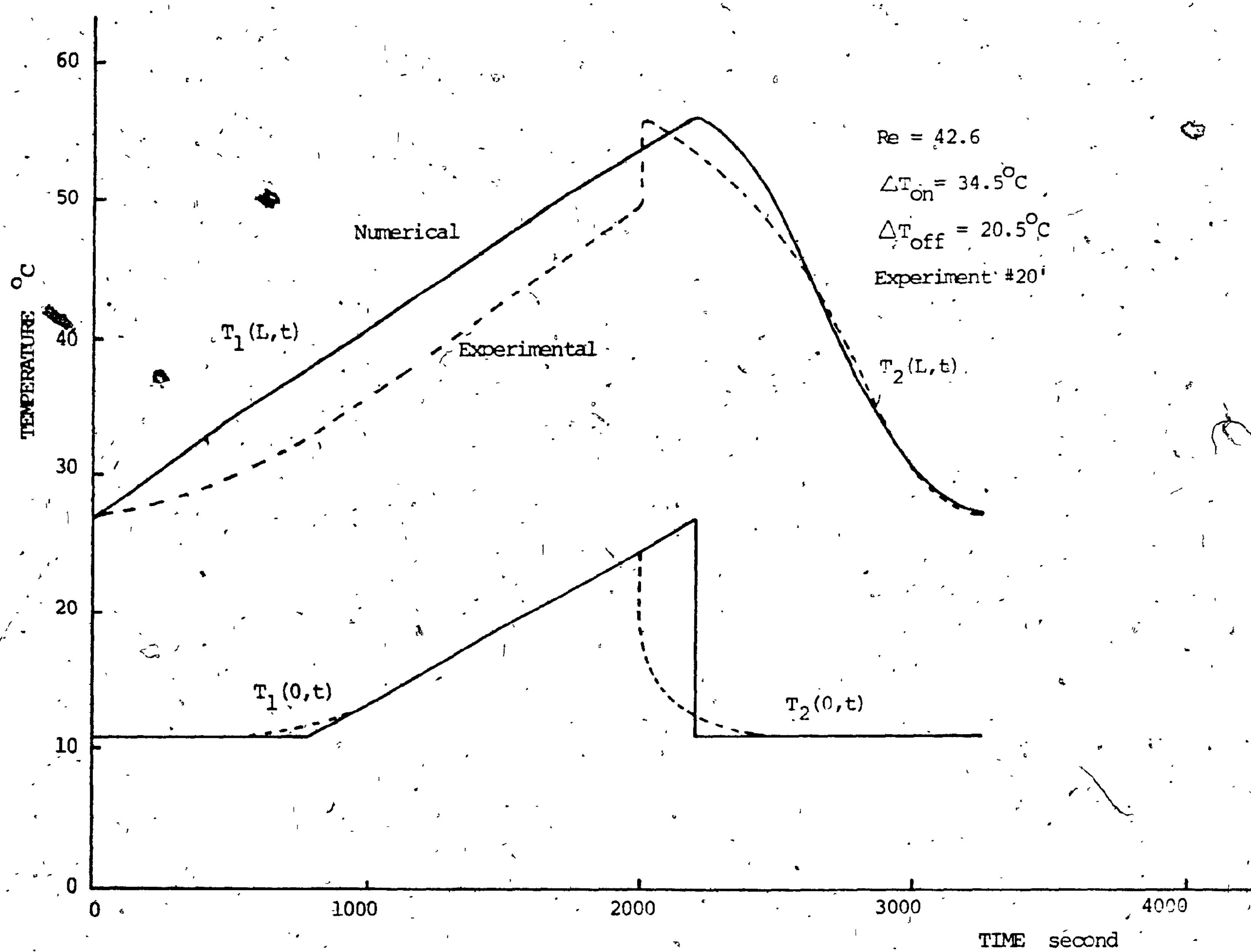

Figure 12. VARIATION OF THE OUTLET WATER TEMPERATURE VS TIME

MARKS ON ORIGINAL

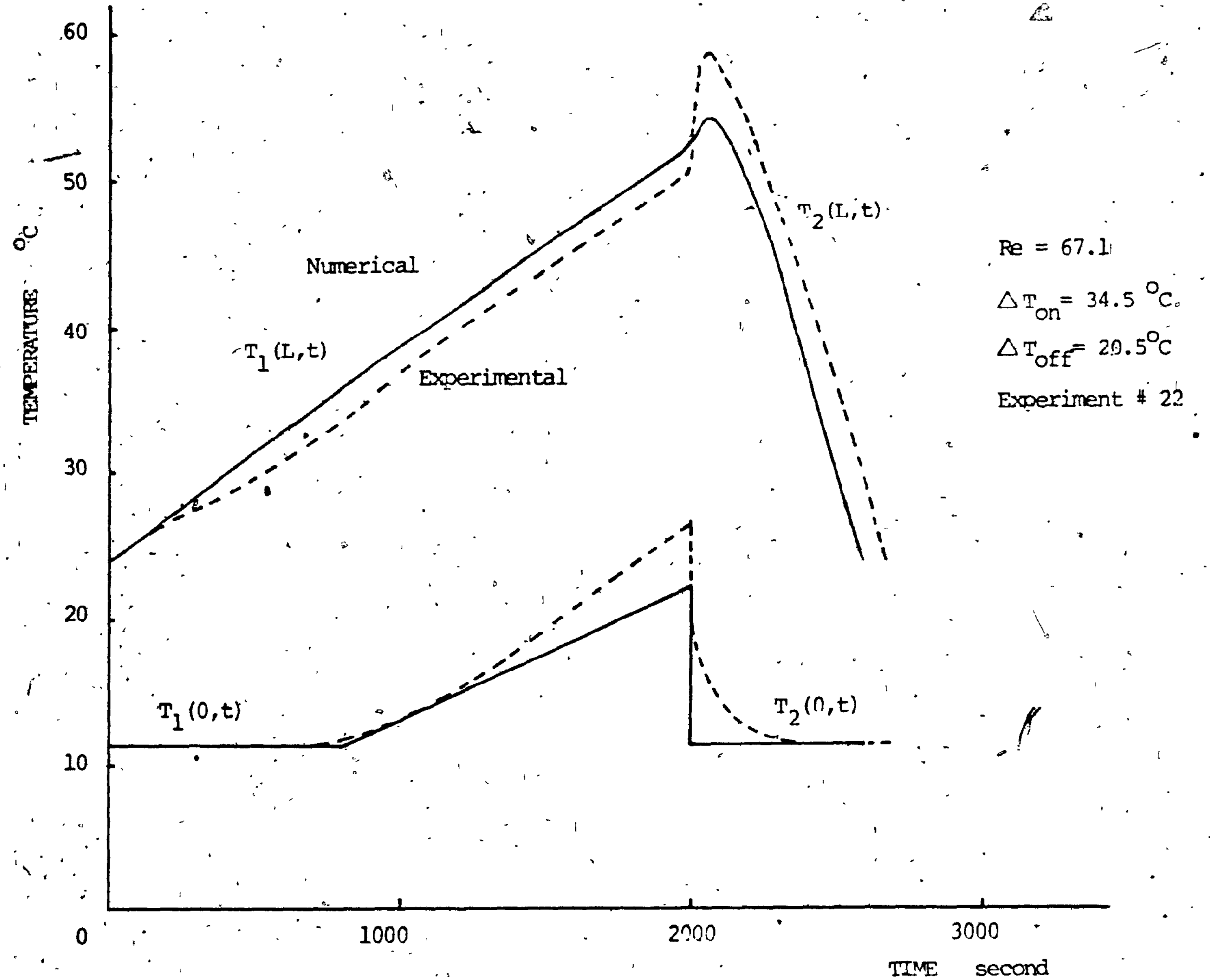

Figure 13 . VARIATION OF THE OUTLET WATER TEMPERATURE VS TIME

MARKS ON ORIGINAL

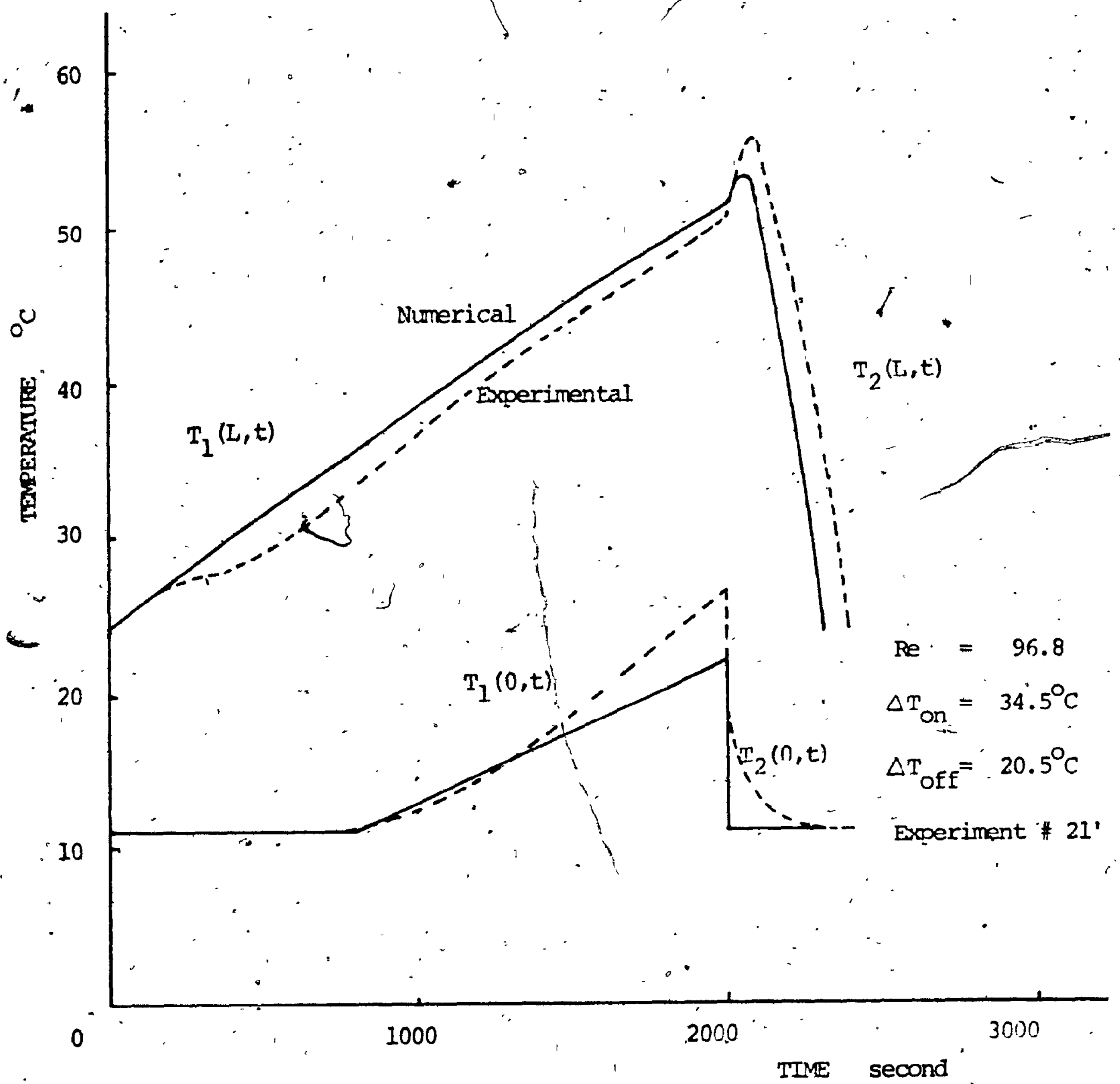

Figure 14. VARIATION OF THE OUTLET WATER TEMPERATURE VS TIME

MARKS ON ORIGINAL

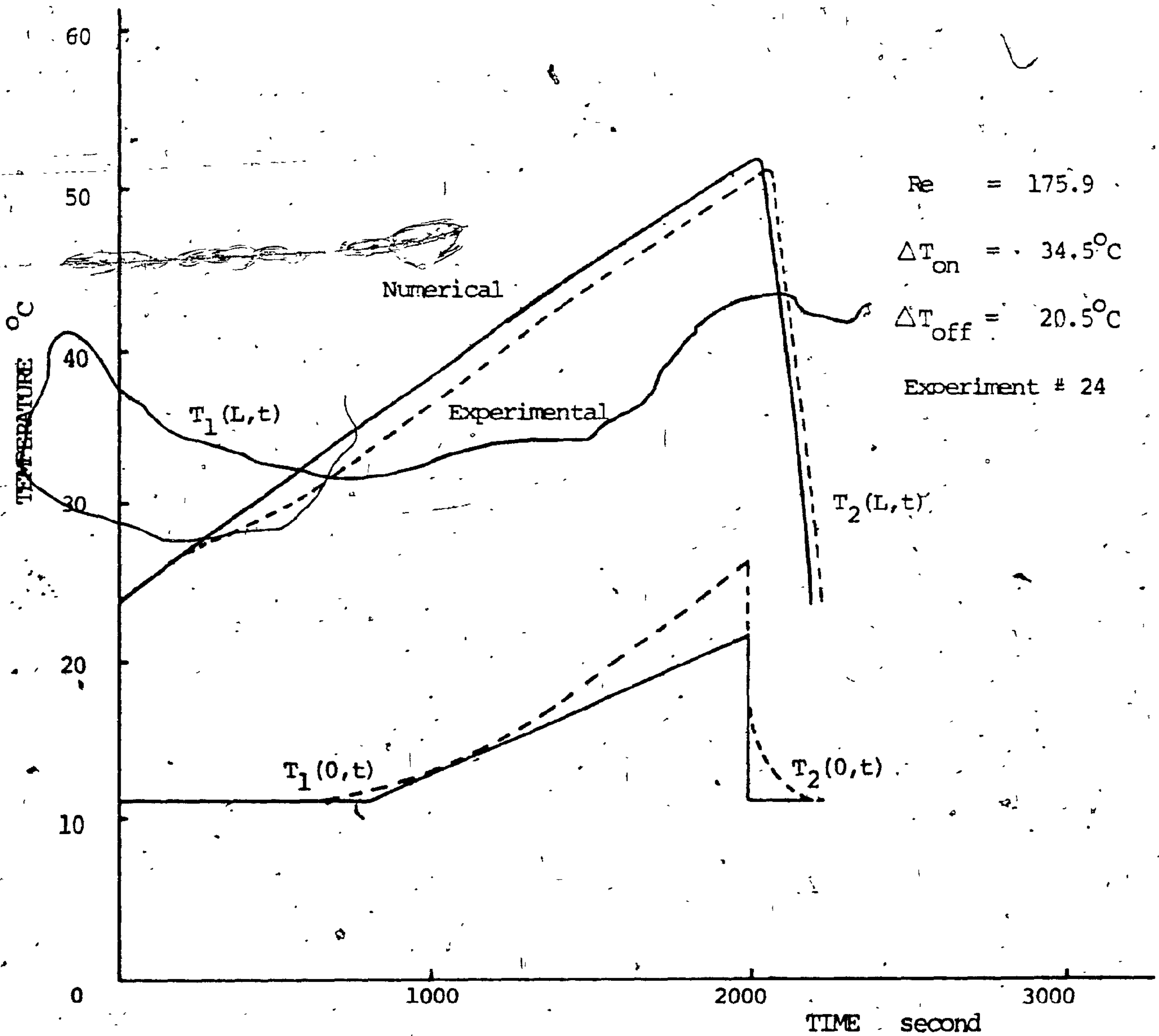

Figure 15. VARIATION OF THE OUTLET WATER TEMPERATURE VS TIME

MARKS ON ORIGINAL

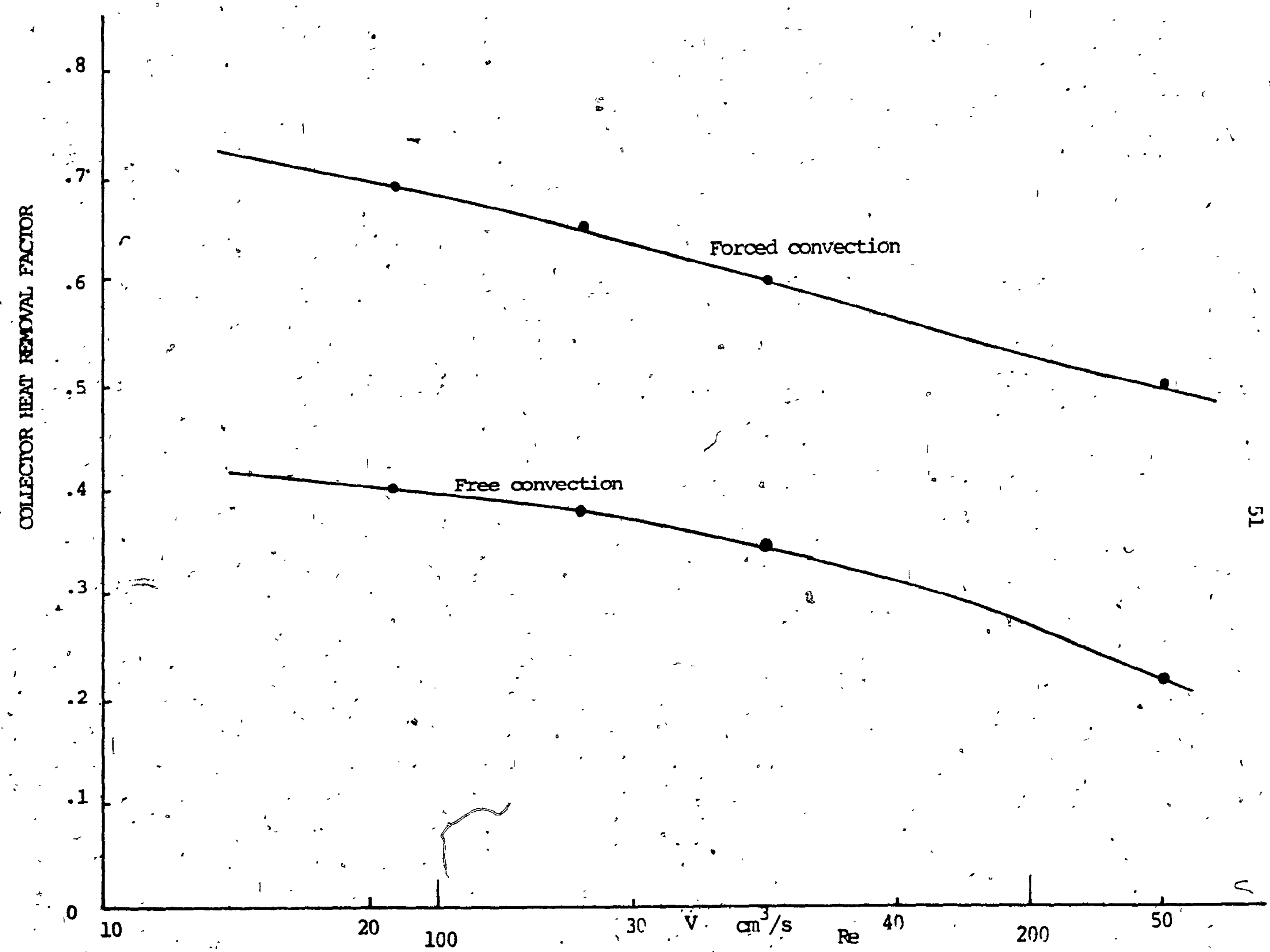

Figure 16. VARIATION OF HEAT REMOVAL FACTOR VS VOLUME FLOW RATE & REYNOLDS NUMBER

MARKS ON ORIGINAL

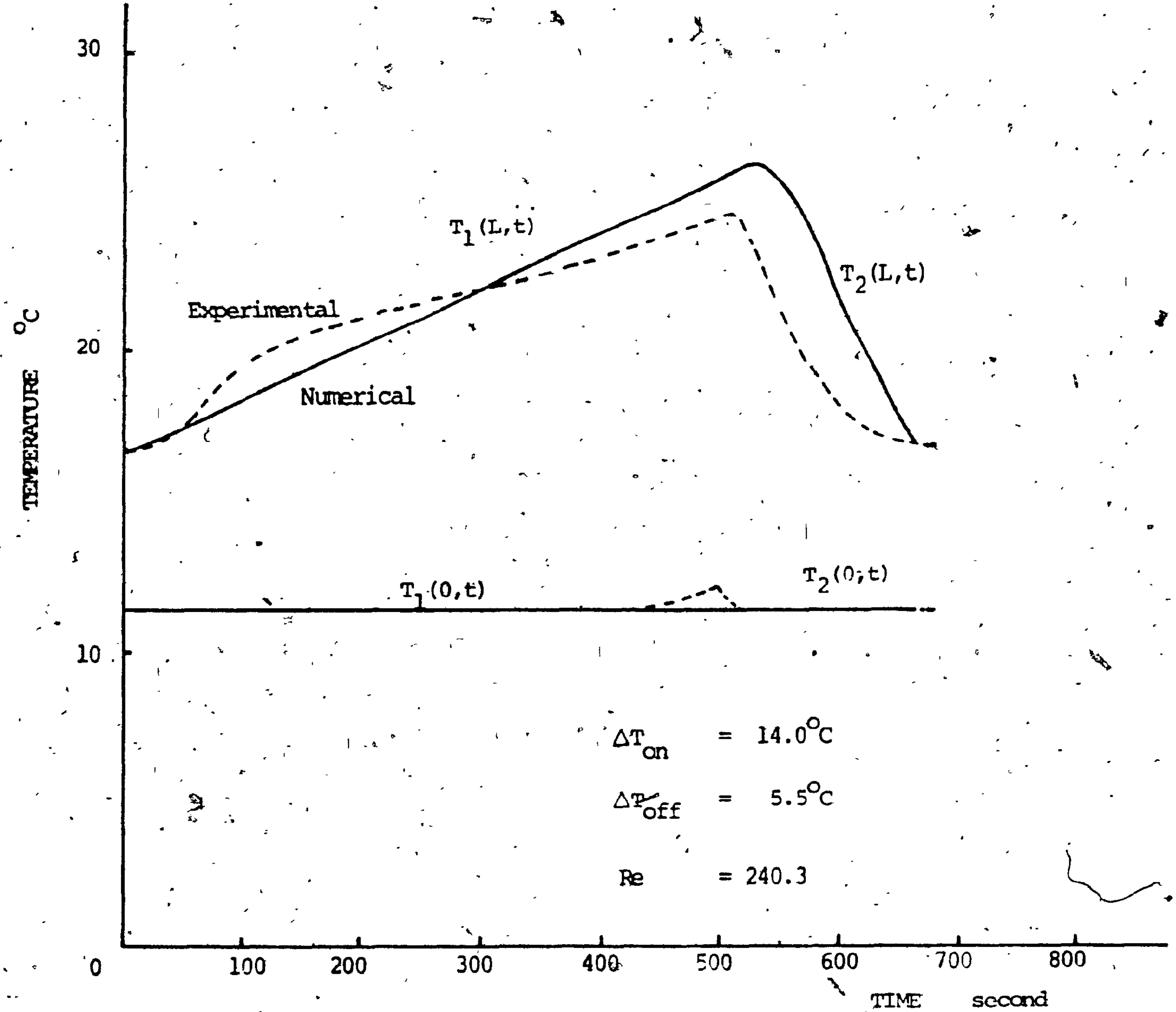

Figure 17. VARIATION OF THE OUTLET WATER TEMPERATURE VS TIME

MARKS ON ORIGINAL

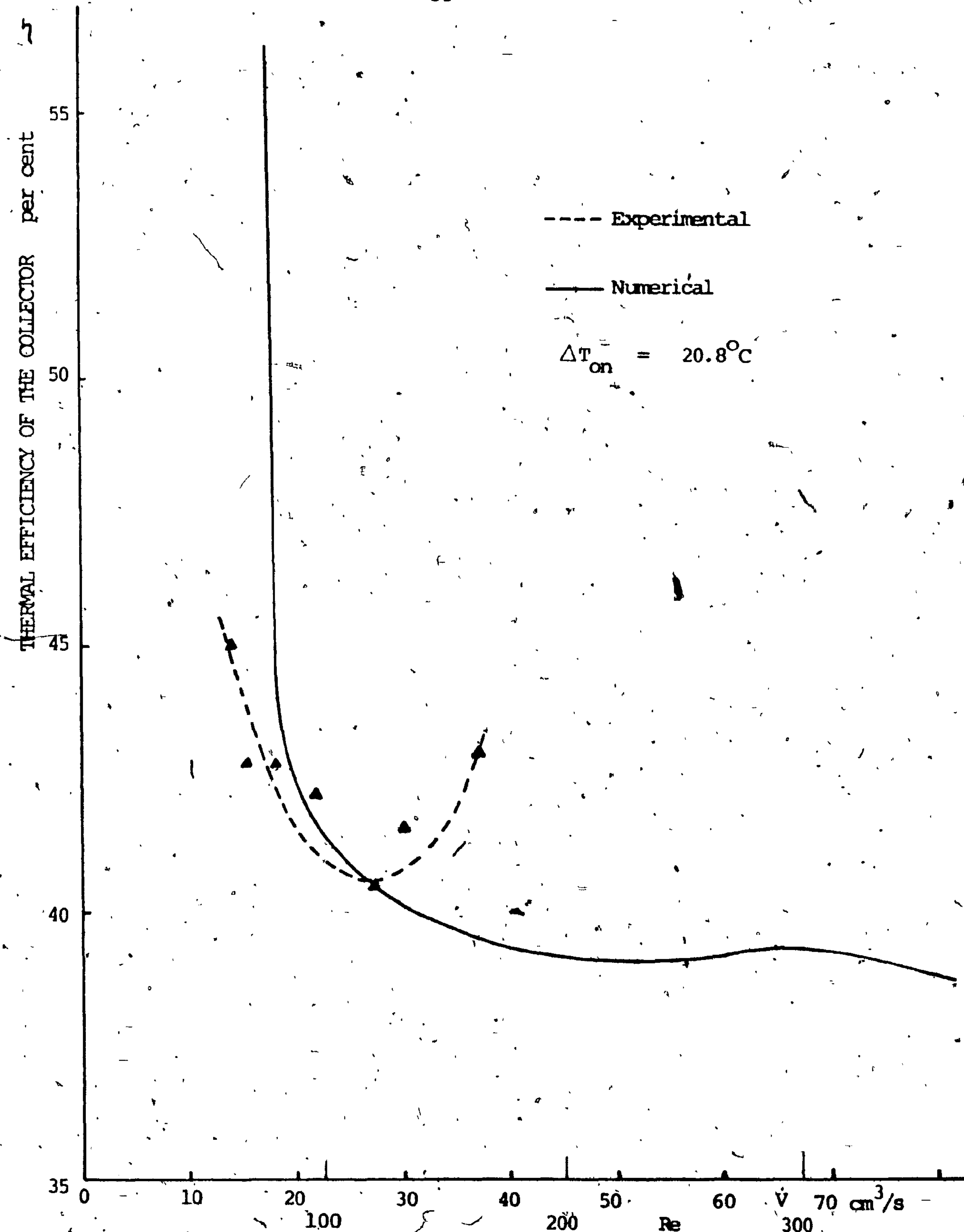

Figure 18. VARIATION OF EFFICIENCY VS VOLUME FLOW RATE & RE NUMBER

MARKS ON ORIGINAL

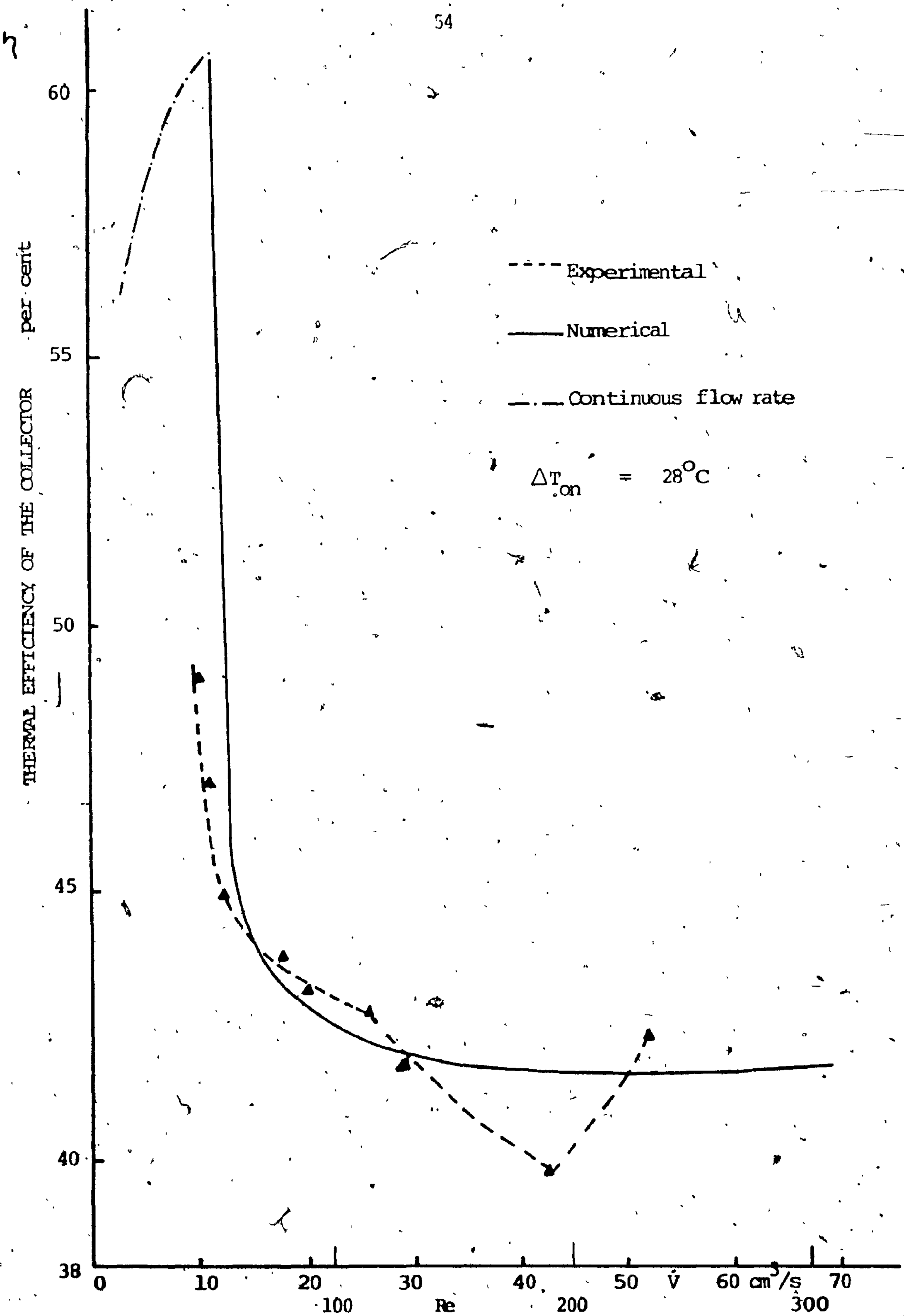

Figure 19. VARIATION OF EFFICIENCY VS VOLUME FLOW RATE & RE NUMBER

MARKS ON ORIGINAL

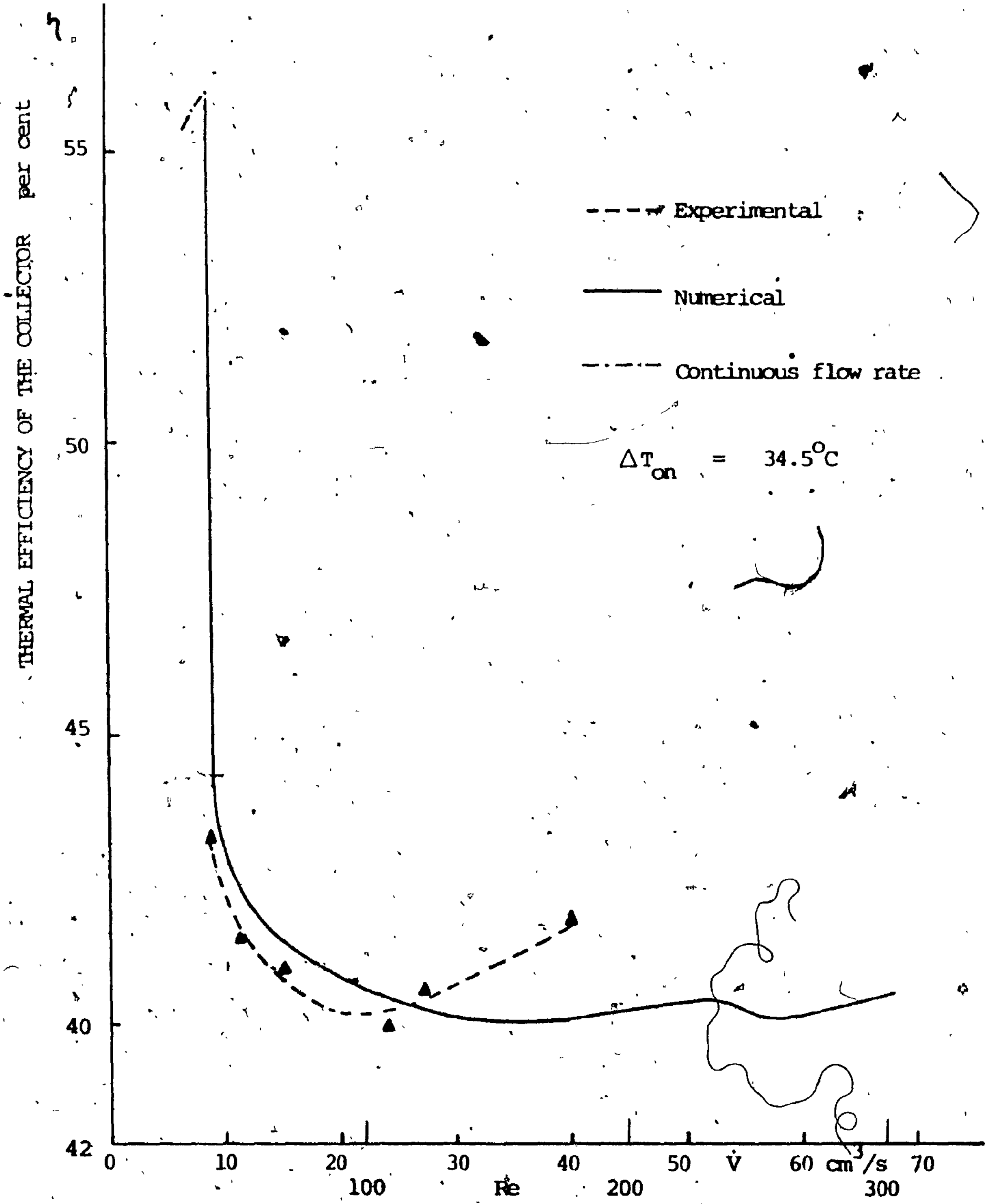

Figure 20. VARIATION OF EFFICIENCY VS VOLUME FLOW RATE & RE NUMBER

MARKS ON ORIGINAL

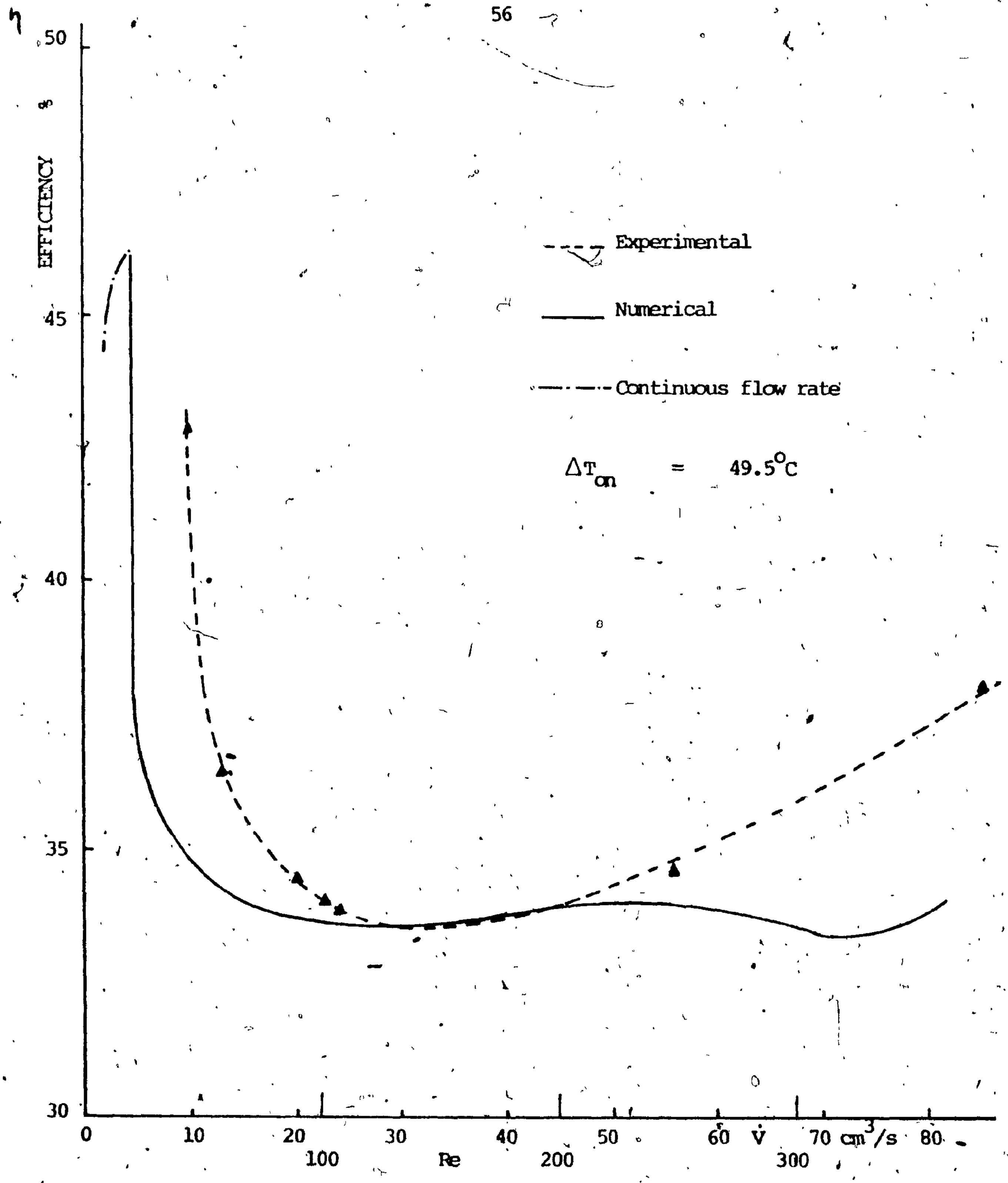

Figure 21. VARIATION OF EFFICIENCY VS VOLUME FLOW RATE & RE NUMBER

MARKS ON ORIGINAL

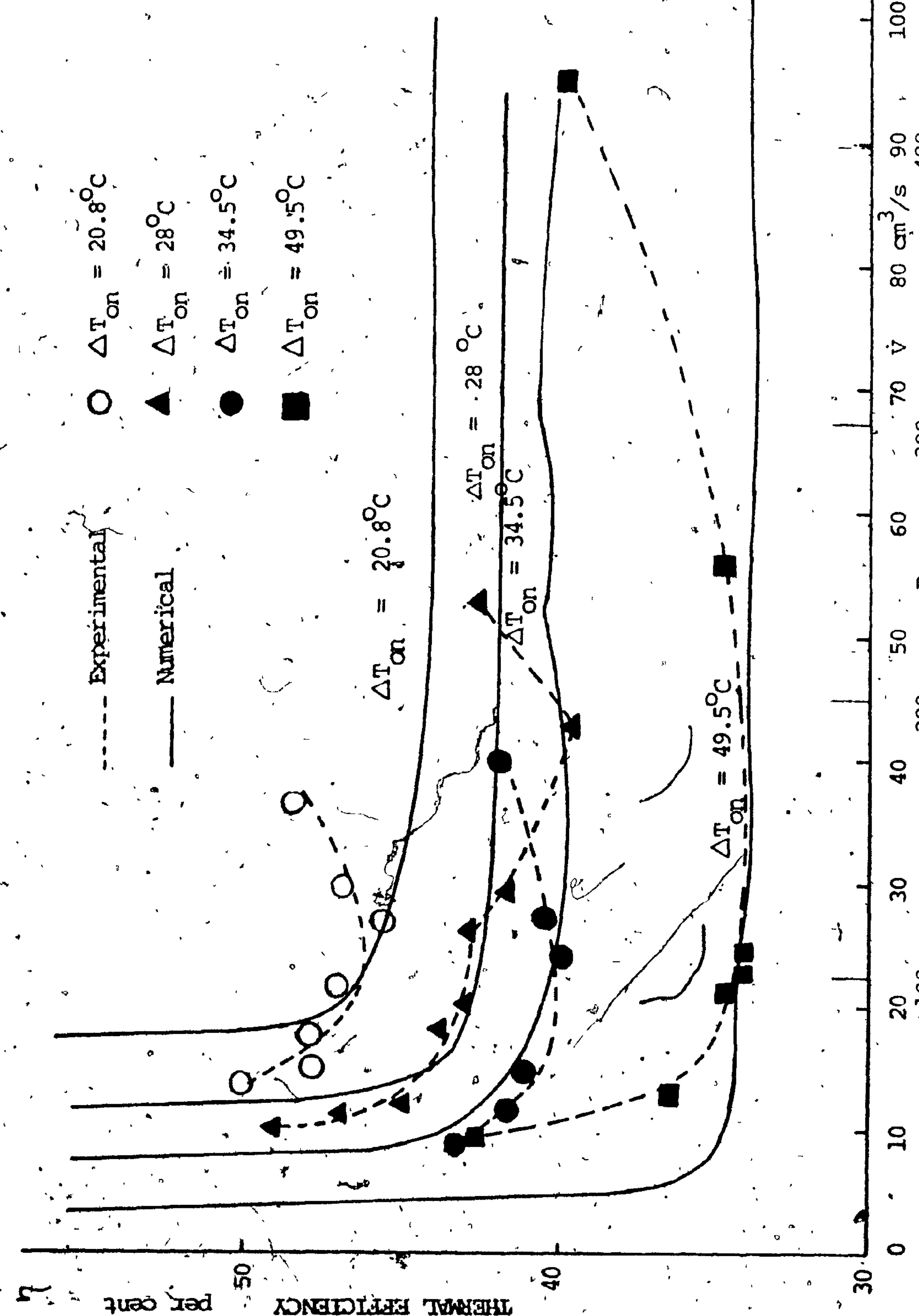

Figure 22. VARIATION OF EFFICIENCY VS VOLUME FLOW RATE & REYNOLDS NUMBER

MARKS ON ORIGINAL

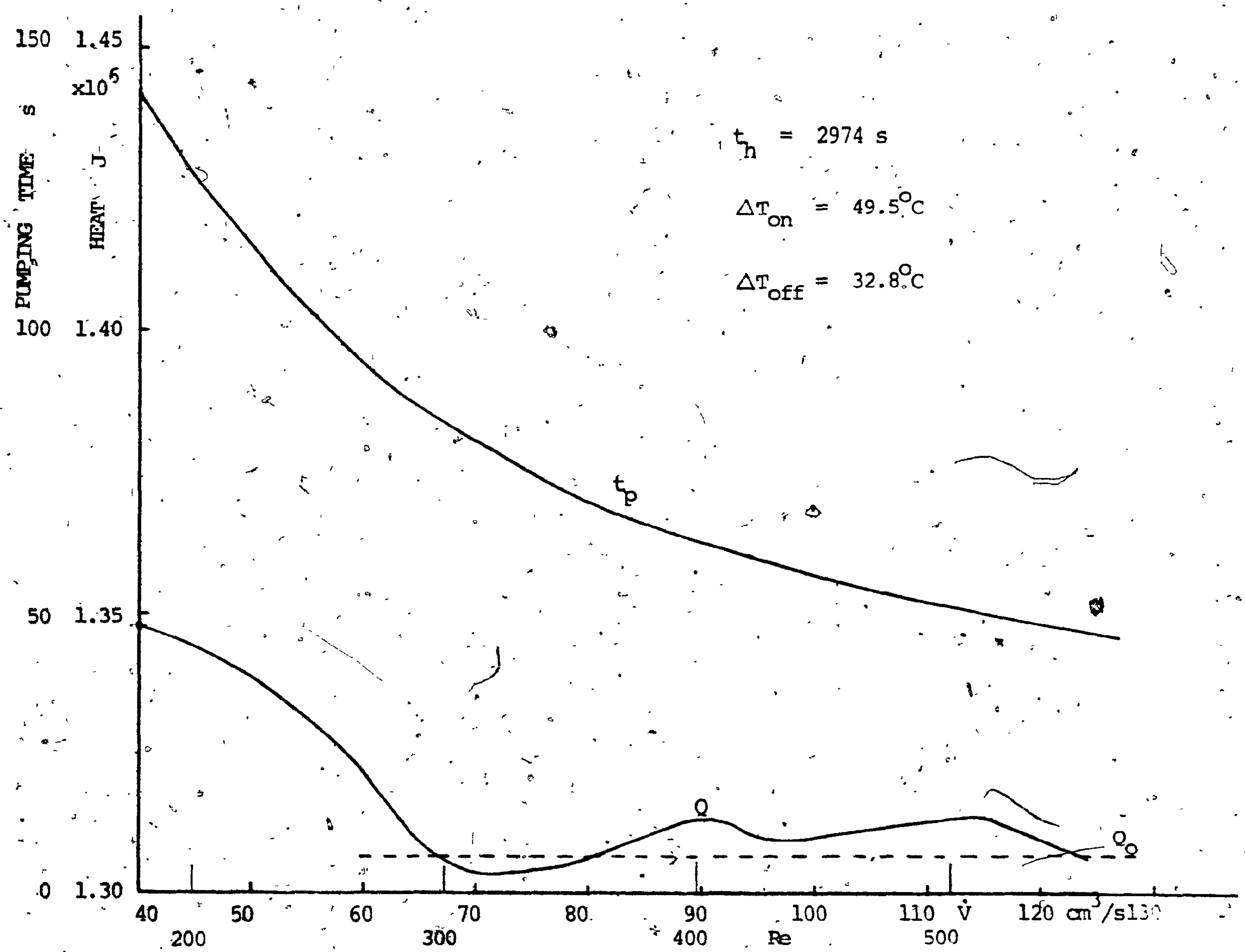

Figure 23. VARIATION OF ABSORBED HEAT PER PERIOD AND PUMPING TIME VS VOLUME FLOW RATE & Re

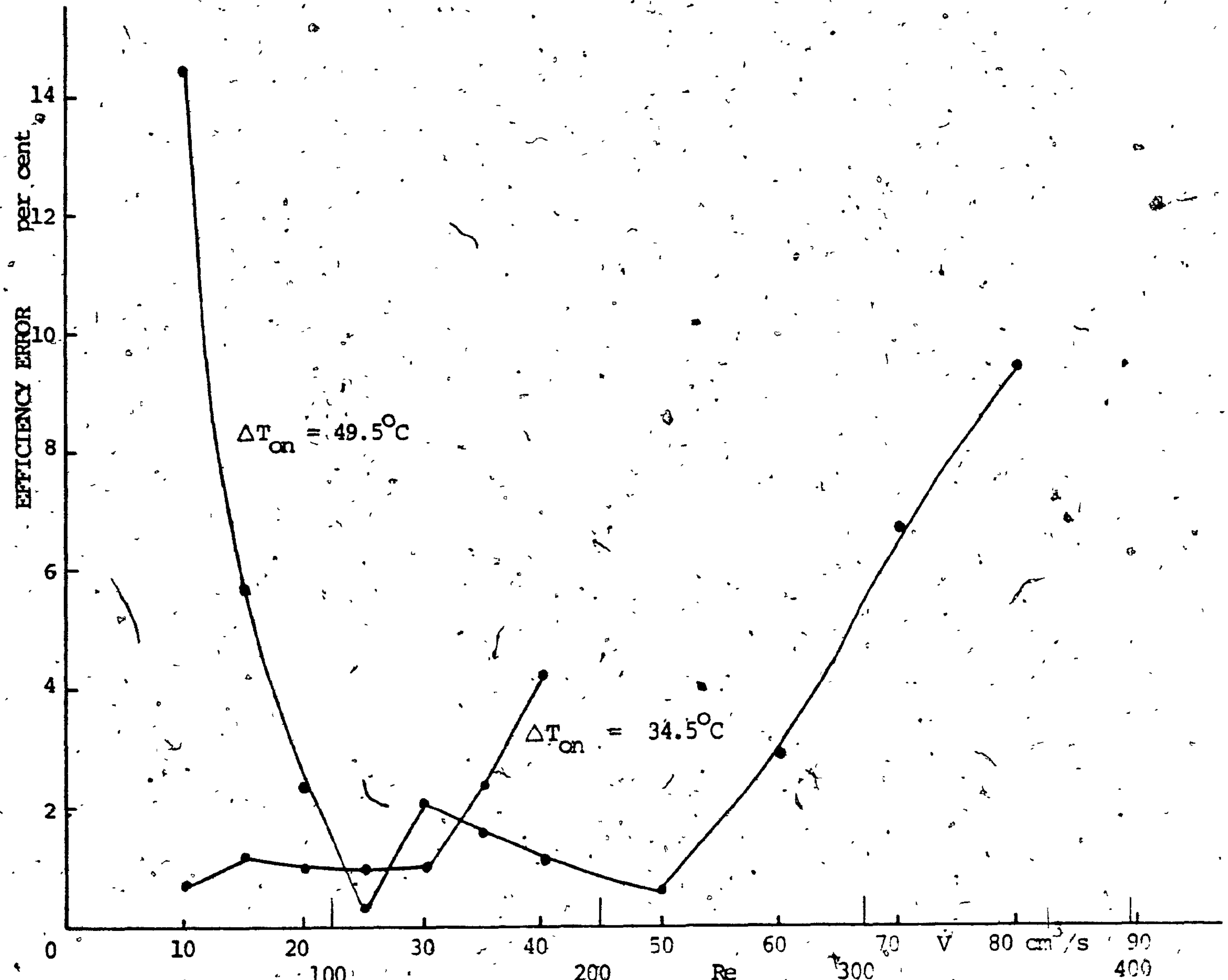

Figure 24. VARIATION OF EFFICIENCY ERROR VS VOLUME FLOW RATE & REYNOLDS NUMBER

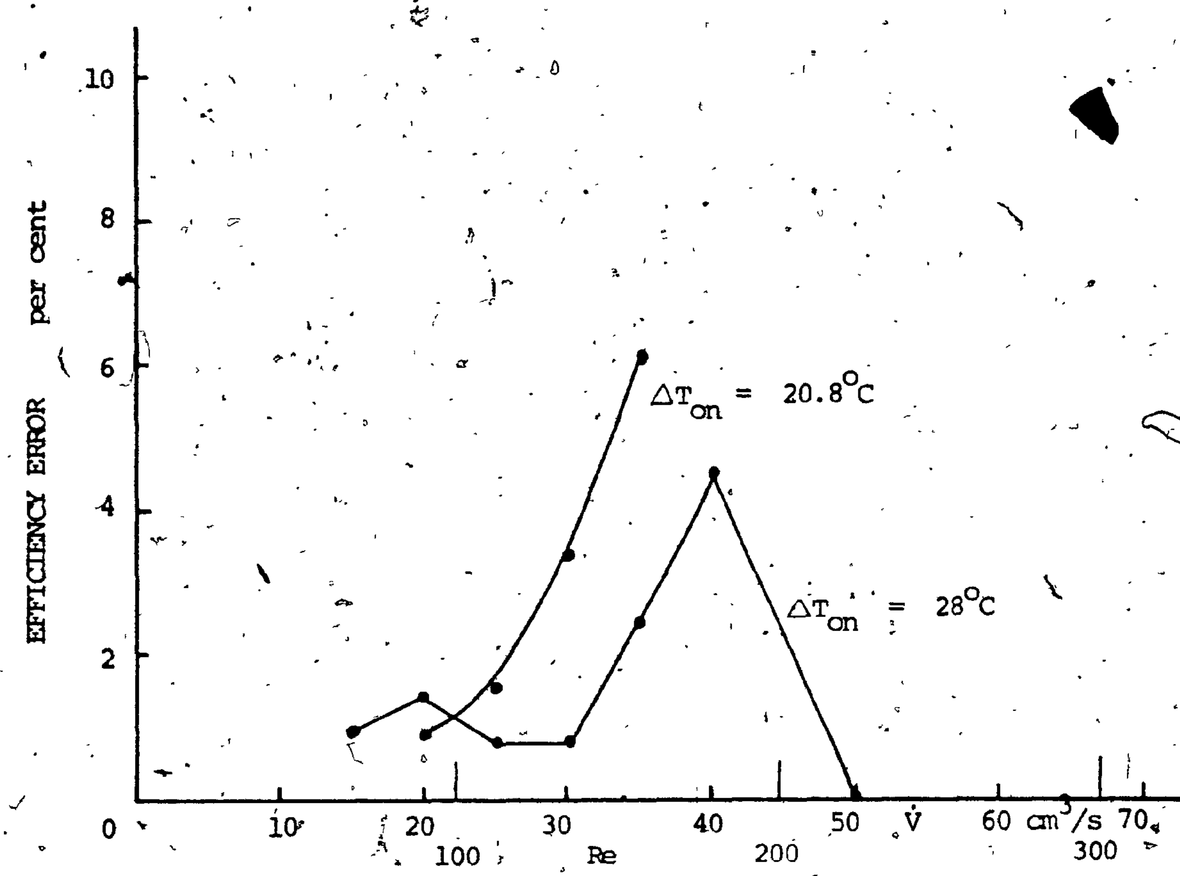

Figure 25. VARIATION OF EFFICIENCY ERROR VS VOLUME FLOW RATE & REYNOLDS NUMBER

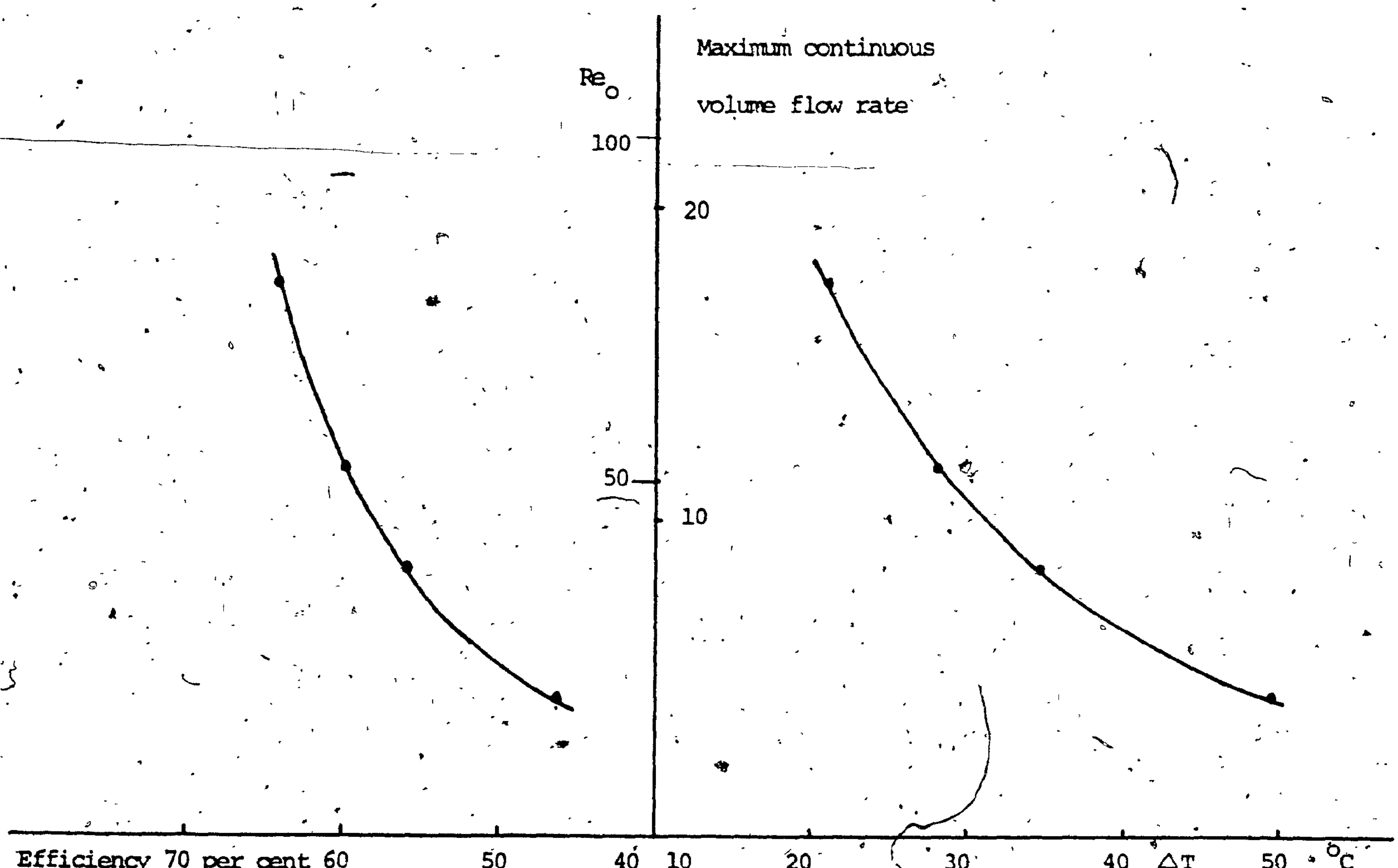

Figure 26. VARIATION OF MAXIMUM EFFICIENCY AND SETTING TEMPERATURE VS MAX. CONTINUOUS FLOW RATE & Re_o